REQUEST

IT IS EARNESTLY DESIRED THAT THE BOOK BE HANDLED WITH CARE AND BE NOT MARKED, UNDERLINED OR DISFIGURED IN ANY OTHER WAY, OTHERWISE IT WILL HAVE TO BE REPLACED OR PAID FOR BY THE BORROWER IN THE INTEREST OF THE LIBRARY.

LIBRARIAN

Outdated Book

A TEXT-BOOK

OF

APPLIED MECHANICS

AND

MECHANICAL ENGINEERING.

IN FIVE VOLUMES.

VOLUME IV.

HYDRAULICS, HYDRAULIC AND REFRIGERATING MACHINERY.

ELEVENTH EDITION.

VENTURI METER

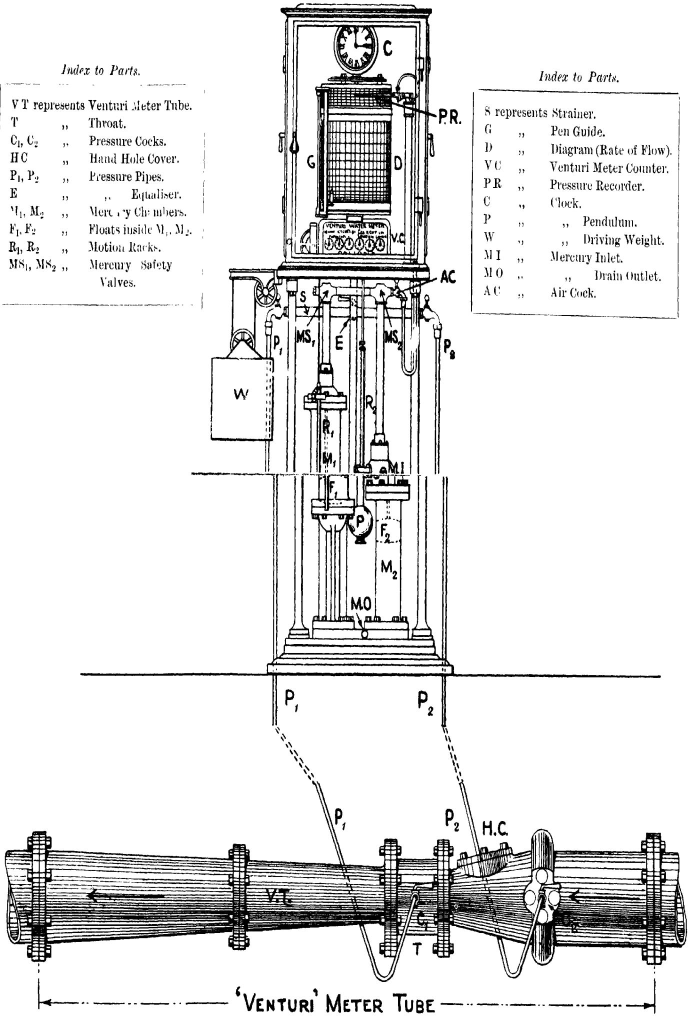
COMBINED DIAGRAM & COUNTER RECORDER.

Index to Parts.

V T	represents	Venturi Meter Tube.
T	,,	Throat.
C_1, C_2	,,	Pressure Cocks.
H C	,,	Hand Hole Cover.
P_1, P_2	,,	Pressure Pipes.
E	,,	,, Equaliser.
M_1, M_2	,,	Mercury Chambers.
F_1, F_2	,,	Floats inside M_1, M_2.
R_1, R_2	,,	Motion Racks.
MS_1, MS_2	,,	Mercury Safety Valves.

Index to Parts.

S	represents	Strainer.
G	,,	Pen Guide.
D	,,	Diagram (Rate of Flow).
V C	,,	Venturi Meter Counter.
P R	,,	Pressure Recorder.
C	,,	Clock.
P	,,	,, Pendulum.
W	,,	,, Driving Weight.
M I	,,	Mercury Inlet.
M O	,,	,, Drain Outlet.
A C	,,	Air Cock.



THE ABOVE RECORDER IS PATENTED BY WALTER G KENT, MANAGING DIRECTOR, MESSRS. GEORGE KENT, LTD., THE MAKERS, HIGH HOLBORN, LONDON.

(See Lecture IV. of Professor Jamieson's Vol. IV. on "Hydraulics" for Description.)

A TEXT-BOOK

OF

APPLIED MECHANICS

AND MECHANICAL ENGINEERING.

Specially Arranged
For the Use of Engineers Qualifying for the Institution of Civil Engineers, The Institution of Mechanical Engineers, The Institution of Electrical Engineers, The Diplomas and Degrees of Technical Colleges and Universities, and Honours Certificates of the City and Guilds of London Institute, in Mechanical Engineering.

BY

ANDREW JAMIESON, M.INST.C.E.,

FORMERLY PROFESSOR OF ENGINEERING IN THE GLASGOW AND WEST OF SCOTLAND TECHNICAL COLLEGE.

VOLUME IV.—HYDRAULICS, HYDRAULIC AND REFRIGERATING MACHINERY.

ELEVENTH EDITION.

NINTH EDITION REVISED BY

EWART S. ANDREWS, B.SC.ENG.

With Numerous Diagrams, Special Plates, and Examination Questions.

LONDON:

CHARLES GRIFFIN AND COMPANY, LIMITED;

42 DRURY LANE, W.C. 2.

1929.

Printed in Great Britain
by Bell & Bain, Limited, Glasgow.

PREFACE

TO THE NINTH EDITION.

ALTHOUGH no alteration has been made in the method of treatment adopted by the late Professor Jamieson in the present volume, considerable revision has been made in the detail arrangement of the book, particularly in the incorporation in their normal place of portions previously appearing in Appendices and in Notes at the end of Chapters.

The most recent papers of the Associate Membership Examination of the Institution of Civil Engineers and of the Mechanical Engineering Examination of the City and Guilds Institute have been included.

It is hoped that the book, in its revised form, will be found of great assistance to students and engineers.

[*Reprinted for Tenth cnd Eleventh Editions.*]

PREFACE

TO THE SEVENTH EDITION.

It has been found necessary to still further subdivide this wide and all-important subject of Advanced Applied Mechanics and Mechanical Engineering.

In order to do so with the least departure and derangement of the previous volumes and editions, it has been advisable and convenient to follow the recent subdivision of this subject as stated in the "Rules and Syllabus of Examinations applying to the Election of Associate Members of The Institution of Civil Engineers."

Moreover, this particular method of subdivision is practised by several Universities and Technical Colleges. It is also being advocated by Teachers in connection with the Boards of Education, and, to a certain extent, by those connected with the City and Guilds of London Examinations in Mechanical Engineering.

Consequently, Volume I. will deal with "Applied Mechanics" proper, Volume II. will discuss and give practical illustrations of "Strength and Elasticity of Materials," Volume III. will be confined to "The Theory of Structures," Volume IV. to "Hydraulics, Hydraulic and Refrigerating Machinery," whilst Volume V. will be greatly enlarged, and treat upon "The Theory of Machines."

Separate Contents and Index have been carefully arranged for each Volume. These enable students to find the details and pages where the different subjects are treated. The Author's system of Engineering Symbols, Abbreviations, and Index Letters have been printed at the beginning of each volume.

It is thus hoped, that the size and cost of each volume will suit the requirements of every Student of Engineering.

The Author has again to thank his chief assistant, Mr. John Ramsay, A.M.Inst.C.E., for his help, and his Publishers for the special care with which they have prepared these five volumes. The Author is specially indebted to Professor Alexander MacLay, B.Sc., C.E., and the Technical Publishing Company, Ltd., Manchester, for permission to abstract certain parts and figures from his excellent book on "*Loci* in Mechanical Drawing, including Point Paths in Mechanisms." These additions appear in Vol. V. upon the "Theory of Machines," along with the new Lectures dealing with the "Kinematics of Machinery."

The Author will feel much obliged to Engineers, Teachers, his B.Sc. and C.E. "Correspondence Students," also to any other Students of Engineering, for hints which may tend to further enhance the attractiveness and usefulness of these works.

ANDREW JAMIESON.

Consulting Engineer and Electrician,
16 ROSSLYN TERRACE, KELVINSIDE,
GLASGOW.

INSTRUCTIONS FOR ANSWERING HOME EXERCISES.

1. Use ordinary foolscap paper, and write on the left side *only*, leaving the facing page blank for corrections and remarks.

2. Put the date of the Exercises at the left-hand top corner; your Name and Address in full, the name of the Subject or Section, as well as number of Lecture or Exercise, in the centre of the first page. The number of each page should be put in the right-hand top corner.

3. Leave a margin 1¼ inches wide on the left-hand side of each page, and in this margin place the *number* of the question *and nothing more*. Also, leave a clear space of *at least* 2 inches deep between your answers.

4. Be sure you understand *exactly* what the question requires you to answer, then give *all* it requires, but *no more*. If unable to fully answer any question, write down your own best attempt and state your difficulties.

5. Make your answers concise, clear, and exact, and always accompany them, if possible, by an *illustrative sketch*. Try to give (1) Side View, (2) Plan, (3) End View. Where asked, or advisable, give Sections, or Half Outside Views and Half-Sections for (1), (2), and (3).

6. Make all sketches large, open, and in the centre of the page. Do not crowd any writing about them. Simply print sizes and index letters (or names of parts), with a bold Sub-heading of what each figure or set of figures represent.

7. Reasonable and easily intelligible contractions (*e.g.*, mathematical, mechanical or electrical, and chemical symbols) are permitted.

CONTENTS.

HYDRAULICS, HYDRAULIC AND REFRIGERATING MACHINERY, WITH PNEUMATIC TOOLS.

PAGES

LECTURE I.

Hydrostatics—Hydraulic Machines.

Hydraulics—Fluids—Viscosity—Transmission of Pressure by a Fluid—Pressure of a Heavy Fluid—Head—Pressure on an Immersed Surface—Examples I., II., III., and IV.—Centre of Pressure—Centre of Pressure on a Rectangle—Triangle—Circle—Example V.—Energy of Still Water—Common Suction Pump—Belt-driven Suction Pump—Example VI.—Air Pump—Single-acting Force Pump—Single-acting Force Pump with Ball Valves—Force Pump with Air Vessel—Continuous Delivery Pumps without Air Vessels—Double-acting Force Pump—Double-acting Circulating Pump—Worthington Steam Pump—Pulsometer Pumps—Roots' Blower—Bramah's Hydraulic Press—Examples VII. and VIII.—Hydraulic Flanging Press—Hydraulic Jack—Examples IX., X., and XI.—Hydraulic Bear—Lead-covering Cable Press—Hydraulic Accumulator—Example XII.—Hydraulic Cranes—Hydraulic Wall Crane—Movable Jigger Crane—Double-Power Hydraulic Crane—Hydraulic Capstan—Questions, ... 1-61

PAGES

LECTURE II.

Efficiency of Machines.

Frictional Resistances and Efficiencies of Machines in General—Example I.—Application to the Steam Engine—Efficiency of a Reversible Machine — Example II. — Movable Hydraulic Cranes — Movable Electric Crane — Details of Hoisting Brake and Levers for Working the 3-ton Electric Crane—Abstract of Report, Tables and Curves on Comparative Trials for Efficiency of 3-ton Hydraulic and Electric Cranes—Relative Cost of Hydraulic and Electric Power—Crane Tests—Explanation of Efficiency Curves—Electric Cranes for Manchester Ship Canal—Questions, . 62-93

LECTURE III.

Bernouilli's Theorem.

Energy of Flowing Water—Bernouilli's Theorem—Jet pumps—Injectors and Ejectors — Hydraulic Ram — Example—Venturi Law and Meter—Theory of the Venturi Tube—Effect of Varying the Throat Ratio—Uses of the Venturi Meter—Kent's Recorder—Friction Curve—Questions, . 94-112

LECTURE IV.

Flow of Water, &c.

Velocity of Efflux and Flow of Water from a Tank—Rectangular Gauge Notch—Thomson's Triangular Notch—Measurement of Head—Measurement of Large Streams—Horse-Power of a Stream—Petot Tube—Free and Forced Vortex—Centrifugal Pressure—Reaction of a Jet—Re-entrant Orifice—Flow in Pipes—Hydraulic Mean Depth—Questions—Solutions to I.C.E. Questions, ... 113-159

LECTURE V.

Water-Wheels and Turbines.

Hydraulic Motors — Overshot Water-Wheel — Breast-Wheels—Undershot Water-Wheel — Fairbairn's Improvements — Clack Mill — Pelton Wheel — Turbines — Girard Turbine

PAGES

—Jonval Turbine—Günther's Governor—Thomson's Vortex Turbine—Little Giant Turbine—Hercules Mixed-Flow Turbine—Centrifugal Pumps and Fans—Efficiency of Pumps and Turbines—Questions, ... 160-197

LECTURE VI.

Refrigerating Machinery.

Refrigeration—Preliminary Considerations—Carbon Dioxide as a Refrigerating Agent—Elementary Refrigerating Apparatus—Simple Refrigerating Machine—Carbon Dioxide Refrigerating Plant—Anhydrous Ammonia as a Refrigerating Agent—De La Vergne's Refrigerating Plant—De La Vergne's Double-acting Compressor—The Linde System of Refrigeration—Apparatus for Transmitting the Cold produced to the Chambers requiring Refrigeration, . . 198-216

APPENDIX.

Useful Constants; Logarithm Tables; Functions of Angles, . 217-222

A.M.I.C.E. Syllabus and Recent Examination Papers, . . . 223-246

City and Guilds Regulations for Mechanical Engineering and Recent Examination Papers, ... 247-254

C.G.S. System of Units, ... 255-258

INDEX, ... 259-263

CONTENTS OF VOLUMES.

VOL. I.—Applied Mechanics.—The Principle of Work and its Applications; Friction, Power Tests, with Efficiencies of Machines.—Velocity and Acceleration. —Motion and Energy.—Energy of Rotation and Centrifugal Force. (Twelfth Edition.)

VOL. II.—Strength of Materials.—Stress, Strain, Elasticity, Factors of Safety, Resilience, Cylinders, ***Shafts, Beams and Girders, Testing Machines, and*** **Testing of Materials of Construction.** (Tenth Edition.)

VOL. III.—Theory of Structures and Graphic Statics, with Applications to Roofs, Cranes, Beams, Girders, and Bridges. (Eleventh Edition.)

VOL. IV.—Hydraulics.—Hydraulic and Refrigerating Machinery. (Eleventh Edition.)

VOL. V.—Theory of Machines.—Loci and Point Paths. —Kinematic Pairs, Links, Chains, &c.—Cranks and Parallel Motions.—Chains and Peaucellier Mechanisms, &c.—Kinematics, Centres and Relative Velocities.—Miscellaneous Mechanisms.—Reversing and Return Motions, &c.—Efficiency of Machines.— Wheel Gearing and Electric Driving.—Friction and Wedge Gearing, with Power Transmitted.—Teeth of Wheels.—Cycloidal Teeth.—Involute Teeth— Bevel and Mortice Wheels, &c.—Friction and Strength of Teeth in Gearing.—Belt, Rope, and Chain Gearing.—Velocity-Ratio and Friction of, with Horse-power Transmitted by Belt and Rope Gearing.—Inertia Forces of Moving Parts and Crank Effort Diagrams of Reciprocating Engines. —Governors and Centrifugals. (Tenth Edition.)

MECHANICAL ENGINEERING SYMBOLS, ABBREVIATIONS, AND INDEX LETTERS

USED IN VOLUMES I. TO V.

OF PROFESSOR JAMIESON'S "APPLIED MECHANICS."

Prefatory Note.—It is very tantalising, as well as a great inconvenience to Students and Engineers, to find so many different symbol letters and terms used for denoting one and the same thing by various writers on mechanics. It is a pity, that British Civil and Mechanical Engineers have not as yet *standardised* their symbols and nomenclature as Chemists and Electrical Engineers have done. The Committee on Notation of the Chamber of Delegates to the International Electrical Congress, which met at Chicago in 1893, recommended a set of "Symbols for Physical Quantities and Abbreviations for Units," which have ever since been (almost) universally adopted throughout the world by Electricians.* This at once enables the results of certain new or corroborative investigations and formulæ, which may have been made and printed anywhere, to be clearly understood anywhere else, without having to specially interpret the precise meaning of each symbol letter.

In the following list of symbols, abbreviations and index letters, the *first* letter of the chief noun or most important word has been used to indicate the same. Where it appeared necessary, the *first* letter or letters of the adjectival substantive or qualifying words have been added, either as a following or as a subscript or suffix letter or letters. For certain specific quantities, ratios, coefficients and angles, small Greek letters have been used, and I have added to this list the complete Greek alphabet, since it may be refreshing to the memory of some to again see and read the names of these letters, which were no doubt quite familiar to them when at school.

* These "Symbols for Physical Quantities and Abbreviations for Units" will be found printed *in full* in the form of a table at the commencement of Munro and Jamieson's *Pocket-Book of Electrical Rules and Tables.* If a similar recommendation were authorised by a committee composed of delegates from the chief Engineering Institutions, it would be gladly adopted by "The Profession" in the same way that the present work of "The Engineering Standards Committee" is being accepted.

TABLE OF MECHANICAL ENGINEERING QUANTITIES, SYMBOLS, UNITS AND THEIR ABBREVIATIONS.

(As used in Vols. I. to V. of Prof. Jamieson's "Applied Mechanics.")

Quantities.	Symbols.	Defining Equations.	Practical Units.	Abbreviations of the Practical Units.
FUNDAMENTAL.				
Length, . . .	L, l	...	Yard, . . . Foot, . . . Inch, . . .	yd. ft. in.
Mass, . . .	M, m	...	Pound, . . .	lb.
Time, . . .	T, t	...	Second, . . Minute, . . Hour, . . .	s. m. h.
GEOMETRIC.				
Surface, . . .	S, s	$S = L^2$	Square foot, . . Square inch, . .	sq. ft. sq. in.
Volume, . . .	V	$V = L^3$	Cubic foot, . . Cubic inch, . .	cb ft. cb. in.
Angle, $\angle$ . .	$\left\{\begin{matrix} \alpha, \beta \\ \theta, \phi \end{matrix}\right\}$	$\alpha = \frac{\text{arc}}{\text{radius}}$	Degree, . . Minute, . . Second, . . Radian $= \frac{180°}{\pi}$.	1° 1′ 1″ rn.
MECHANICAL.				
Velocity, . .	v	$v = \frac{L}{T}$	Foot per second, .	$\frac{\text{ft.}}{\text{s.}}$
Angular velocity, .	ω	$\omega = \frac{v}{L} = \frac{\theta}{t}$	Revs. per second, . Revs. per minute, Radians per second,	r.p.s. r.p.m. ω
Acceleration, . .	a, g	$a = \frac{v}{T}$	Foot per sec. per sec.	$\frac{\text{ft.}}{\text{s}^2}$
Force, . . .	F, f	...	Pound weight (gravitational unit),	lb. wt. (or lb.)
	W, w	$F = M a$	Poundal (absolute unit),	pdl.
Pressure (per unit area),	p	$p = \frac{F}{s}$	Pound per sq. inch,	lb. □″
Work, . . .	$(W h)$	$W h = F L$	Foot-pound, . .	ft.-lb.
Potential energy, .	E_P	$E_P = W h$	Foot-pound, . .	ft.-lb.
Kinetic energy, .	E_K	$E_K = \frac{W v^2}{2 g}$	Foot-pound, . .	ft.-lb.
Power or activity, .	HP	$\text{H.P.} = \frac{W h}{T}$	Horse power, . Ft.-lb. per min., . Ft.-lb. per sec., .	H.P. ft.-lb./m. ft.-lb./s.
Moment of inertia, .	I	$I = M k^2$	... ...	lb.-ft.^{2}
Density, . . .	ρ	$\rho = \frac{M}{V}$	Pound per cb. ft., . Pound per cb. in., .	$\frac{\text{lb.}}{\text{ft.}^3}$ $\frac{\text{lb.}}{\text{in.}^3}$

OTHER SYMBOLS AND ABBREVIATIONS IN VOLS. I. TO V.

A for Areas.
B, b ,, Breadths.
C, c, k ,, Constants, ratios.
c.g. ,, Centre of gravity.
D, d ,, Diameters depths, deflections.
D_1, D_2, D_3 ,, Drivers in gearing.
E ,, Modulus of elasticity.
e ,, Velocity ratio in wheel gearing.
F_1, F_2, F_3 ,, Followers in gearing.
f_s, f_t ,, Forces of shear and tension.
H, h ,, Heights, heads.
H.P., h.p. ,, Horse-power.
B.H.P. ,, Brake horse-power.
E.H.P. ,, Effective ,,
I.H.P. ,, Indicated ,,
k ,, { Radius of gyration, or, Coef. of discharge in hydraulics.
N, n ,, Numbers—*e.g.*, number of revs. per min., number of teeth, &c.
P, Q ,, Push or pull forces.
$R_1 R_2$,, Reactions, resultants, radii, resistances.
s ,, { Seconds, space, surface. Displacement, distance.
S F ,, Shearing force.
T M ,, Torsional moment.
T R ,, Torsional resistance.
B M ,, Bending moment.
M R ,, Moment of resistance.
R M ,, Resisting moment.
T_d, T_s ,, Tensions on driving and slack sides of belts or ropes, &c.
W_L, W_T, W_U ,, Lost, total, and useful work.

x, y, z for Unknown quantities.
Z ,, Modulus of section.
Z_t ,, ,, tension.
Z_c ,, ,, compression.

Δ, δ, d for Differential signs which are prefixed to another letter; then the two together represent a very small quantity.
ϵ, e ,, Represents base of Naperian Logs $=2{\cdot}7182$; for example, $\log_e 3 = 1{\cdot}1$.
η ,, Efficiency.
λ ,, Length ratio of ship to model.
μ ,, Coefficient of friction.
π ,, Circumference of a circle ÷ its diameter.
ρ ,, Radius of curvature, radian.

Σ for Symbol for sum total of a number of quantities.
$\int_0^x$,, Sign of integration or summation between limits 0 and x.
$\sim$,, Sign for the difference between two quantities.
□ ,, Sign for square—*e.g.*, 10 □″ = 10 square inches.
— ,, Sign over two letters, $\overline{PQ}$, for a force acting from P to → Q, means that they represent a vector quantity, which has (1) magnitude, (2) direction, (3) sense.
$\geqq$,, Sign for equal to or greater than.
$\leqq$,, Sign for equal to or less than.
$\doteqdot$,, Approximately equal to.

GREEK ALPHABET.

A	α	Alpha.	I	ι	Iota.	P	ρ	Rho.
B	β	Beta.	K	κ	Kappa.	Σ	σ or ς	Sigma.
Γ	γ	Gamma.	Λ	λ	Lambda	T	τ	Tau.
Δ	δ	Delta.	M	μ	Mu	Υ	υ	Upsilon.
E	ϵ	Epsilon.	N	ν	Nu	Φ	ϕ	Phi.
Z	ζ	Zeta.	Ξ	ξ	Xi.	X	χ	Chi.
H	η	Eta.	O	o	Omicron.	Ψ	ψ	Psi.
Θ	θ	Theta.	Π	π	Pi.	Ω	ω	Omĕga.

VOLUME IV.

ON

HYDRAULICS AND HYDRAULIC MACHINERY.

LECTURE I.

HYDROSTATICS—HYDRAULIC MACHINES.

CONTENTS.—Hydraulics—Fluids—Viscosity—Transmission of Pressure by a Fluid—Pressure of a Heavy Fluid—Head—Pressure on an Immersed Surface—Examples I., II., III., and IV.—Centre of Pressure—Centre of Pressure on a Rectangle—Triangle—Circle—Example V.—Energy of Still Water—Common Suction Pump—Belt-driven Suction Pump—Example VI.—Air Pump—Single-acting Force Pump—Single-acting Force Pump with Ball Valves—Force Pump with Air Vessel—Continuous Delivery Pumps without Air Vessels—Double-acting Force Pump—Double-acting Circulating Pump—Worthington Steam Pump—Pulsometer Pumps—Roots' Blower—Bramah's Hydraulic Press—Examples VII. and VIII.—Hydraulic Flanging Press—Hydraulic Jack—Examples IX., X., and XI.—Hydraulic Bear—Lead-covering Cable Press—Hydraulic Accumulator—Example XII.—Hydraulic Cranes—Hydraulic Wall Crane—Movable Jigger Crane—Double Power Hydraulic Crane—Hydraulic Capstan—Questions.

Hydraulics.—In its widest sense, the term "Hydraulics" is given to the study of the mechanical properties of fluids and their application to practical purposes. In a more restricted sense, it refers to the science of the pressure and flow of water and their applications in engineering. It is divided into two sections:—**Hydrostatics**, the science of fluids at rest; and **Hydrokinetics**, the science of fluids in motion.

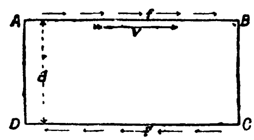
Fluids.—In many investigations it is necessary for simplicity to assume that we are dealing with a *perfect fluid;* that is, one which possesses the following property:—

DEFINITION.—**A fluid is a substance which offers no resistance to a continuous change of shape.**

There are two kinds of fluids—those which are practically incompressible, termed *liquids;* and those which are easily compressed, called *gases* and *vapours.* We know of no substance which completely fulfils the above definition; but water, many other liquids, and all gases, so nearly comply with it, that for many purposes we may, in practice, consider them as perfect fluids.

Viscosity.—Ordinary fluids, however, do offer some resistance to a change of shape, and the property in virtue of which they do so is called the *viscosity* of the fluid. A substance, such as syrup, which offers considerable resistance to a rapid change of form, but which goes on changing its shape so long as any deforming forces are applied to it, however small these forces may be, is usually called a *viscous fluid*. This term, strictly speaking, applies to all fluids, since all have some viscosity. It should, however, be noted that even the most mobile fluid will offer an appreciable resistance to a *sudden* deformation, because parts of it have to be set in motion and their inertia comes into play. This must not be confounded with their viscosity.

A solid body differs from a viscous fluid in that a small force produces in it a definite change of shape in a short time, and thereafter no further deformation takes place. Many solids, however, such as lead, tin, copper, and iron, when subjected to very great stresses, behave like viscous fluids, and keep *flowing* as long as the pressure is kept up. Even with very small forces, such apparently solid bodies as sealing wax and cobbler's wax, which fly to pieces when we subject them to a sudden force, such as a blow from a hammer, will gradually yield when sufficient time is allowed, and consequently they must be considered as very viscous fluids. For instance, a leaden bullet will sink in a thick piece of cobbler's wax and a cork will rise upwards through it, just as they would do through syrup or water, but they may take many months or years to do so.



VISCOSITY OF A FLUID.

The viscosity of a fluid is measured by the shear stress required to deform it at the uniform rate of unit shear strain per unit time. Thus, *if the figure represents a small portion of fluid and if a* tangential stress f acts along A B and C D, the fluid will change its shape by the part A P moving along with a velocity v, relatively to the part D C.

$$\text{Then,}\quad \left.\begin{array}{l}\textit{Shear strain produced}\\ \quad\textit{in unit time}\end{array}\right\} = \frac{\textit{Velocity of A}}{\textit{Distance A D}}$$

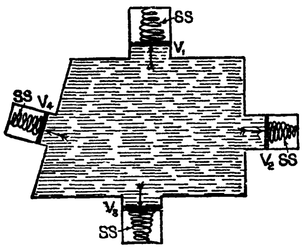
$$\text{Or,}\quad \textit{Rate of shear} = \frac{v}{d} = \omega.$$

$$\therefore \quad \text{Coefficient of viscosity} = \mu = \frac{f}{\omega}. \quad \ldots \ldots \quad \text{(I)}$$

When dealing with fluids at rest and in hydraulic machines,

such as presses, cranes, &c., in which the motion of the liquid is comparatively slow, we need not take account of their viscosity; but when considering their flow through pipes and channels it becomes of great importance.

Transmission of Pressure by a Fluid.—Pascal's law, that "fluids transmit pressure equally in all directions," follows at once from our definition of a fluid. Thus, take a vessel filled with a fluid and fitted with several frictionless pistons which all have the same area and are held in place by springs. If we now apply an inward force through the spring to one of the pistons, say V_1, we shall find that each of the other pistons will be pushed outward with the same force. Had the pistons been of different areas we should have found the forces proportional to their areas, showing that the pressure per unit area is the same in all directions.

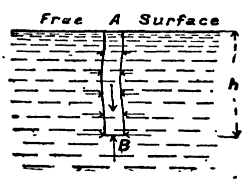


(HORIZONTAL SECTION.)

TRANSMISSION OF PRESSURE BY FLUIDS.

Another property following from our definition is that the pressure on any surface, real or imagined, is everywhere normal to that surface.

Pressure of a Heavy Fluid—Head.—Had the fluid in the above experiment been water or mercury and the pistons placed at different levels, we should have found that the pressure was not the same on all of them, but greatest on the lowest and least on the uppermost piston. This difference is due to the weight of the fluid. For example, if we have a quantity of liquid, the pressure at the bottom end B, of a vertical column A B, would be greater than at A; and, since the pressures round the sides of the column balance *one another, the weight and the pressures on the ends must be in equilibrium.* The difference of pressure is, therefore, equal to the weight of a cylinder of liquid whose length is A B and whose cross section has unit area. This will obviously be proportional to the length of the column A B; that is, to the difference of level.



PRESSURE OF A HEAVY FLUID.

In the figure, the upper surface is open to the atmosphere, and is, therefore, called the *Free Surface.* The pressure at A is atmospheric, but in connection with hydraulics it is customary to reckon

this as the zero of reference, and when we speak of the pressure of a fluid we mean the excess of its pressure above that of the atmosphere.

Since we can have any pressure by taking A B of suitable length, we very often measure a pressure by the length of the vertical column of liquid which it will support—or, what is the same thing, which will produce the same pressure—and we shall term this length the *Head*. The column itself we shall refer to as the *Pressure Column*. With water each foot of head, and with mercury each inch, corresponds to nearly half a pound per square inch. The exact figures are:—

1 Foot of Water	=	0·434 lb. per square inch.
1 Inch of Mercury	=	0·491 ,, ,,

If h be the height of the free surface above the point we are considering, and w the weight of unit volume of the liquid, the pressure per unit area will be:—

$$p = w\,h. \qquad \text{(II)}$$

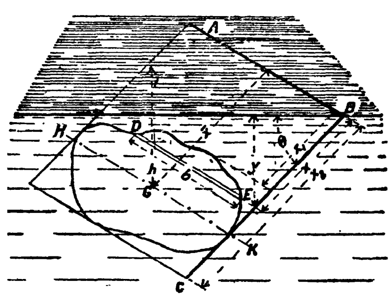
For fresh water, w may be taken as 62·42, or nearly $62\frac{1}{2}$ lbs. (1,000 ounces) per cubic foot, or 0·0361 lb. per cubic inch.

Pressure on an Immersed Surface.—If the above pressure be exerted on every unit of a surface whose area is a, the *total pressure* on that surface will be:—

$$P = a \times p = a\,w\,h. \qquad \text{(III)}$$

This is also frequently called "the pressure," but it is usually quite clear whether the total pressure or the intensity of pressure is meant, although they are commonly denoted by the same term.

On a plane surface which is not level, the intensity of pressure is not the same for all parts. In such a case, we may find the total pressure as follows:—



PRESSURE ON A SUBMERGED SURFACE.

Let A B be the intersection of the plane of the submerged surface with the free surface of the water. Draw the line B C in that plane perpendicular to A B, and take D E, a very narrow strip or element of the surface, at right angles to B C

and distant x from A B. A B and D E will be parallel since they are in one plane and both perpendicular to B C. Consequently, D E will be horizontal, and, therefore, the intensity of pressure over it will be uniform and equal to $w\,y$, y being the depth of the strip below the surface. Hence, if b be the breadth D E of the surface (*i.e.*, the length of the strip), and $d\,x$ that of the strip, the total pressure on the element will be $b\,d\,x \times w\,y$, or $b\,d\,x\,w\,x \sin\theta$ since $y = x \sin\theta$, if θ be the inclination of the plane to the horizontal. Now, we can split up the whole surface into a very large number of such elements and the total pressure on it will be the sum of all those on the elements :—

$$\therefore \qquad P = w \sin\theta \int_{x_1}^{x_2} b\,x\,d\,x.$$

But $b\,x\,d\,x$ is the area of an element multiplied by its distance from A B, and, therefore, from the definition of the centre of gravity of a lamina, the integral is equal to the whole area a multiplied by the distance of its centre from A B. Let this distance be $\bar{x}$, and let $h = \bar{x} \sin\theta$, be the depth of the centre below the surface :—

$$\text{Then,} \qquad P = w \sin\theta \times a\,\bar{x} = a\,w\,h. \qquad . \quad . \quad . \quad (\text{III}_a)$$

This is evidently the same pressure as if the surface were level and immersed at the same depth as its centre of gravity.

Example I.—A cylindrical tank, 6 feet in diameter and 10 feet deep, is filled with water; find the bursting pressure round the base of the tank, and the pressure on its base.

Answer.—The bursting pressure round the base is measured by the intensity of the fluid pressure on any small area of the curved surface infinitely near to the base. This pressure will be exactly equal to that on the base. Hence, the question resolves itself into finding the intensity of the pressure on the base.

$\therefore$ Bursting pressure round the base of tank $=$ *Pressure per square inch on base.*

" " $= a\,h\,w.$

" " $= \dfrac{1}{144} \times 10 \times 62{\cdot}5 = 4{\cdot}34$ lbs. per sq. in.

Again, Total pressure on base $=$ *Area of base in sq. ins.* $\times$ *pressure per sq. in.*

Or, " " $= \pi \times 36 \times 36 \times 4{\cdot}34 = 17{,}700$ lbs.

EXAMPLE II.—A circular water tank is 20 feet in diameter and 25 feet deep. It is constructed of 6 rings of cast-iron plates. Find the *total* stress on any vertical section of the bottom row of plates made by a plane passing through the axis of the cylinder, neglecting any assistance afforded by the flanges or connection with the bottom plate.

ANSWER.—It has been proved in Lect. I., Vol. II., that when a cylindrical shell is subjected to internal fluid pressure, the *total* stress in the material along any section made by a plane containing the axis of the cylinder is equal to the total fluid pressure on either side of that part of the plane intercepted within the cylinder.

Hence, total stress in material of bottom row of plates = total fluid pressure on vertical plane through the axis of the cylinder at the bottom row of plates.

Since the breadth of each ring $= \frac{25}{6} = 4\frac{1}{6}$ ft.; therefore, depth of *c. g.* of bottom ring $= h = 25 - \frac{1}{2} \times 4\frac{1}{6} = 22{\cdot}97$ ft.

$$\therefore \left.\begin{array}{l}\text{Total stress in material}\\ \text{along section at bottom}\\ \text{row of plates}\end{array}\right\} = a\,h\,w,$$

$$\text{,,} \qquad \text{,,} \qquad = (20 \times 4\tfrac{1}{6}) \times 22{\cdot}97 \times 62{\cdot}5 \text{ lbs.}$$

$$\text{,,} \qquad \text{,,} \qquad = 119{,}400 \text{ lbs.}$$

EXAMPLE III.—How is the pressure of water on a given area immersed in it ascertained? A water tank, 8 feet long and 8 feet wide, with an inclined base, is 12 feet deep at the front and 6 feet deep at the back, and is filled with water. Find the pressure in lbs. on each of the four sides, and on the base; water weighing $62\frac{1}{2}$ lbs. per cubic foot.

ANSWER.—The total fluid pressure on any area immersed in the fluid is given by the formula—$P = a\,h\,w$.

Where a = Area of surface exposed to the fluid pressure,

h = Depth of centre of gravity of immersed area below free surface of fluid,

w = Weight of a cubic unit of fluid.

The shape and dimensions of the tank will be readily seen from the figure.

(*a*) *To find the total pressure on the front* A B C D.

Here, $a = AD \times DC = 8 \times 12 = 96$ sq. ft.

$h = \frac{1}{2}$ depth D C $= 6$ ft. $\qquad w = 62\frac{1}{2}$ lbs. per cubic ft.

$\therefore$ Pressure on front A B C D $= a\,h\,w$,

$$\text{,,} \qquad \text{,,} \qquad = 96 \times 6 \times 62\tfrac{1}{2} = 36{,}000 \text{ lbs.}$$

(*b*) *To find the total pressure on the back* E F M N.

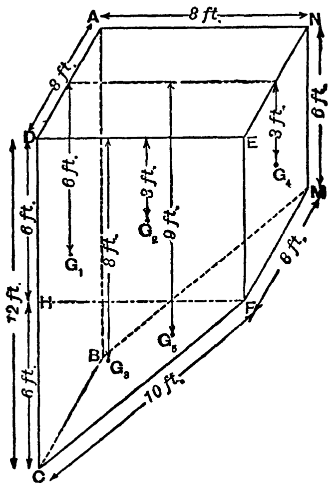
Here, $a = \text{F M} \times \text{M N} = 8 \times 6 = 48$ sq. ft.; $h = \frac{1}{2}\,\text{E F} = 3$ ft.

$\therefore$ **Pressure on back E F M N** $= a\,h\,w$,

„ „ $= 48 \times 3 \times 62\frac{1}{2} =$ **9,000 lbs.**

(*c*) *To find the total pressure on base* C B M F.

Before we can find the area of the base, we must know its length C F. From F draw F H parallel to E D, and therefore perpendicular to D C. Then C H F is a right-angled triangle



PRESSURE ON SIDES OF TANK.

whose sides are F H = E D = 8 ft., and H C = D C − D H = D C − E F = 6 ft.

$$\therefore \qquad \text{C F} = \sqrt{\text{H F}^2 + \text{H C}^2} = \sqrt{8^2 + 6^2} = 10 \text{ ft.}$$

$$\therefore \qquad a = \text{C F} \times \text{C B} = 10 \times 8 = 80 \text{ sq. ft.}$$

Again, the depth of the *c. g.* of the base C B M F is clearly—

$$h = \tfrac{1}{2}\,(\text{D C} + \text{E F}) = 9 \text{ ft.}$$

$\therefore$ **Pressure on base C B M F** $= a\,h\,w$,

„ „ $= 80 \times 9 \times 62\frac{1}{2} =$ **45,000 lbs.**

(*d*) *To find the total pressure on either side* C D E F or A B M N.

In this case it is perhaps best to divide the trapezoidal area C D E F into two figures whose centres of gravity can be easily determined. Thus, the line F H divides the side C D E F into a rectangle D E F H, and a triangle F H C. Then the total pressure on C D E F is equal to the sum of the pressure on D E F H and F H C.

Area of D E F H = 8 × 6 = 48 sq. ft.

And, *Depth of c. g. of area* D E F H $= \frac{1}{2}$ E F = 3 ft.

∴ *Pressure on* D E F H = $a\,h\,w$

" " $= 48 \times 3 \times 62\frac{1}{2}$ lbs.

Again, *Area of* F H C $= \frac{1}{2}$ H F × H C $= \frac{1}{2} \times 8 \times 6 = 24$ sq. ft.

The *c. g.* of triangle F H C is at a distance of $\frac{1}{3}$ of H C below the horizontal F H, and therefore at a distance of $6 + \frac{1}{3}$ of 6 or 8 feet below D E.

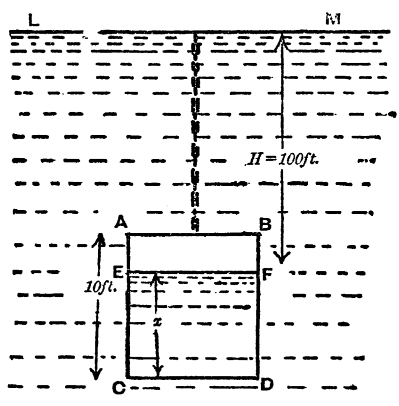
∴ *Pressure on* F H C = $a\,h\,w$,

" " $= 24 \times 8 \times 62\frac{1}{2}$ lbs.

∴ Pressure on side C D E F or A B M N $= 48 \times 3 \times 62\frac{1}{2} + 24 \times 8 \times 62\frac{1}{2}$ lbs.,

" " $= 48 \times 62\frac{1}{2} \times (3 + 4) = 21{,}000$ lbs.

Example IV.—A cylindrical vessel, 10 feet long, open at one



Pressure in Diving Bell.

end and closed at the other, forms a diving bell. It is lowered

into water with its open end downwards until the surface of the water in the cylinder is at a depth of 100 feet. Find how far the water has risen in the cylinder, and the pressure of the contained air. (Take the height of the water barometer as 34 feet.)

ANSWER.—

Let H = Depth of surface of water in bell = 100 feet.
,, h = Height of water barometer = 34 feet.
,, l = Length of cylinder forming bell = 10 feet.
,, x = Height that water rises in bell.

Before the bell is immersed in the water the pressure of the contained air is simply that due to the atmosphere. After immersion the pressure will be greater than that of the atmosphere by an amount due to a head of water of H feet.

Assuming, then, that the air in the bell has been compressed according to Boyle's Law ($p\,v$ = a const.), we get:—

$$\left.\begin{array}{c}\textit{Press. of compressed air}\\ \times\ \textit{Vol. of compressed air}\end{array}\right\} = \left\{\begin{array}{c}\textit{Press. of atmosphere}\\ \times\ \textit{Vol. of bell.}\end{array}\right.$$

$$\therefore \qquad \frac{\textit{Vol. of compressed air}}{\textit{Vol. of bell}} = \frac{\textit{Press. of atmosphere}}{\textit{Press. of compressed air}}.$$

Since the bell is of uniform cross sectional area throughout, we get:—

$$\frac{\textit{Vol. of compressed air}}{\textit{Vol. of bell}} = \frac{l-x}{l}.$$

$$\therefore \qquad \frac{l-x}{l} = \frac{h}{\mathrm{H}+h}.$$

$$\therefore \qquad x = \frac{\mathrm{H}\,l}{\mathrm{H}+h} = \frac{100 \times 10}{100+34} = 7{\cdot}46 \text{ feet.}$$

The pressure of the air in the bell when immersed is equal to the pressure due to a depth of (H + h) feet of water.

$$\therefore \quad \text{Pressure of air in bell} = a\,(\mathrm{H}+h)\,\mathrm{W} = \frac{1}{144} \times 134 \times 62{\cdot}5$$

$$\text{,,} \qquad \text{,,} \qquad = 58{\cdot}16 \text{ lbs. per sq. inch.}$$

Centre of Pressure.—We could balance the pressure on an immersed surface by a single force—the reverse of the resultant of the pressure—acting through a certain point in the plane of the surface, and this point is called the *Centre of Pressure.*

To find the depth of the centre of pressure we may proceed as follows :—

Referring to our former figure we see that the moment about A B of the pressure on the elementary strip D E is $w\,y \times b\,d\,x \times x$ or $w \sin\theta\, b\, x^2\, d\,x$. Hence the total moment is:—

$$M = w \sin\theta \int_{x_1}^{x_2} b\,x^2\,d\,x.$$

Now, $b\,x^2\,d\,x$ is the product of the area of an element into the square of its distance from A B, and consequently $\int_{x_1}^{x_2} b\,x^2\,d\,x$ is the second moment, or moment of inertia, of the area about A B. As shown in equation (III) of Lect. XII., Vol. I., it is, therefore, equal to $I + a\,\bar{x}^2$, where I is the moment of inertia about an axis H K, through the centre of gravity parallel to A B :—

$$\therefore \qquad M = w \sin\theta\,(I + a\,\bar{x}^2).$$

Again, the moment of the resultant must be the sum of the moments of its components. Let X be the distance of the centre of pressure from A B :—

Then, $\qquad P\,X = M = w \sin\theta\,(I + a\,\bar{x}^2).$

$$\therefore \qquad X = \frac{w \sin\theta\,(I + a\,\bar{x}^2)}{P} = \frac{w \sin\theta\,(I + a\,\bar{x}^2)}{w \sin\theta\, a\,\bar{x}}.$$

$$\therefore \qquad X = \frac{I + a\,\bar{x}^2}{a\,\bar{x}}. \qquad \text{(IV)}$$

That is, the distance of the centre of pressure from A B is the ratio of the second moment of the surface about A B to its first moment about the same axis.

If for I we write $a\,k^2$, k being the radius of gyration about the axis H K, and h for $\bar{x} \sin\theta$, we get :—

$$X = \frac{a\,k^2 + a\,\bar{x}^2}{a\,\bar{x}} = \frac{k^2 + \bar{x}^2}{\bar{x}} = \frac{k^2 \sin^2\theta + h^2}{h \sin\theta}. \qquad \text{(V)}$$

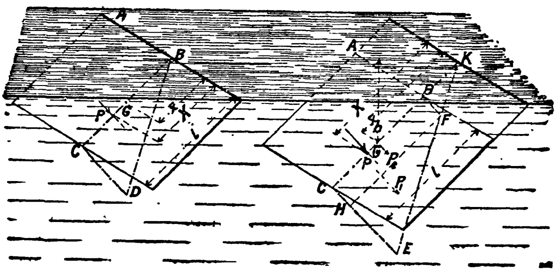
We shall now show how to apply these results to a few simple cases, and will also explain an easier method for special cases.

Centre of Pressure on a Rectangle.—First, consider a rectangle immersed with one edge in the surface. We find from Table II. in Lect. XII., Vol. I., that the value of k^2 for a rectangle is $\frac{1}{12}\,l^2$,

where l is the length of the rectangle at right angles to the axis. Also, $\bar{x}$ will be $\frac{1}{2}l$:—

$$\therefore \qquad X = \frac{\frac{1}{12}l^2 + \frac{1}{4}l^2}{\frac{1}{2}l} = (\tfrac{1}{6} + \tfrac{1}{2})\,l = \tfrac{2}{3}l. \quad . \quad . \quad . \quad (VI)$$

If the rectangle be immersed further, until its centre is at a depth h, the top edge being kept horizontal, we do not get such a simple result, but it can be at once obtained for any given case from equation (V).



Centre of Pressure on a Rectangle.

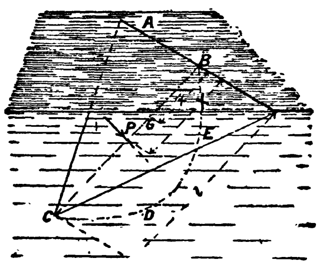
We can also obtain our result in the following manner:—At every point in B C—the central line of the rectangle—draw a line at right angles to it of such a length as to represent the whole pressure on a horizontal strip at that level. When the ends of these are joined we will have a triangle B C D, whose area represents the total pressure on the rectangle. The resultant pressure will pass through the centre of gravity of this triangle, and will, therefore, be two-thirds down from the vertex. Hence, the centre of pressure is distant two-thirds of the length of the rectangle from the top.

When the upper edge of the rectangle is below the surface, we obtain, instead of a triangle to represent the pressure, a trapezium B F E C, whose inclined sides, when produced, meet in the surface of the liquid. We ascertain the centre of pressure by finding the resultant of two forces, P_1 and P_2, the former of which is proportional to the area of the triangle F E H, and is two-thirds down from F, while the latter passes through the centre of the rectangle B F H C, and is proportional to its area. This may be done graphically as explained in Lecture V., Vol. III.

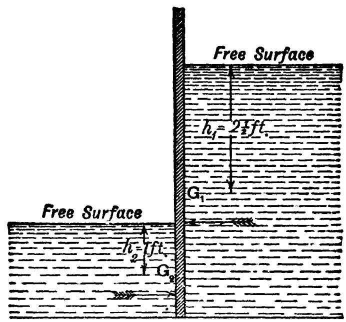
Centre of Pressure on a Triangle.—For a triangle with its base in the surface, $k^2 = \frac{1}{18}l^2$, and $\bar{x} = \frac{1}{3}l$:—

$$\therefore \qquad X = \frac{\frac{1}{18}l^2 + \frac{1}{9}l^2}{\frac{1}{3}l} = (\tfrac{1}{6} + \tfrac{1}{3})\,l = \tfrac{1}{2}l. \quad . \quad . \quad . \quad (VII)$$

This may also be proved geometrically. The intensity of pressure on a horizontal strip is proportional to its depth below the surface, while the length of the strip, and, therefore, its area, is proportional to its distance from the vertex C. Consequently, the whole pressure on each horizontal element will be proportional to the product $x\ (l - x)$ for elements of the same width. The area representing the pressure will, therefore, be a parabola, B E D C, passing through B and C, with its axis perpendicular to B C, and, consequently, the centre of pressure must be half way down.



CENTRE OF PRESSURE ON A TRIANGLE.



PRESSURE ON SLUICE GATE.

If the vertex is in the surface, and the base horizontal, then $\bar{x} = \frac{2}{3}\, l$:—

$$\therefore \qquad X = \frac{\frac{1}{18}\, l^2 + \frac{4}{9}\, l^2}{\frac{2}{3}\, l} = \left(\tfrac{1}{12} + \tfrac{2}{3}\right) l = \tfrac{3}{4}\, l. \quad . \quad . \quad \text{(VIII)}$$

Centre of Pressure on a Circle.—The only other case we shall consider is that of a circle immersed vertically, with its centre at a depth h. Here $k^2 = \frac{1}{4}\, r^2$, and $\bar{x} = h$:—

$$\therefore \qquad X = \frac{\frac{1}{4}\, r^2 + h^2}{h}. \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(IX)}$$

When the circumference just touches the surface, $h = r$, and this becomes :—

$$X = \tfrac{5}{4}\, r = \tfrac{5}{8}\, d. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(X)}$$

Where r is the radius, and d the diameter of the circle.

EXAMPLE V.—A sluice gate is 4 feet broad and 6 feet deep, and the water rises to a height of 5 feet on one side, and 2 feet on the other side. Find the pressure on the gate, and the centres of pressure.

Answer.—The net pressure on the sluice gate is evidently equal to the difference of the pressures on the two sides.

Total Pressure on Back $= a_1 h_1 w = (4 \times 5) \times 2{\cdot}5 \times 62{\cdot}5 = 3{,}125$ lbs.

" " *Front* $= a_2 h_2 w = (4 \times 2) \times 1 \times 62{\cdot}5 = 500$ "

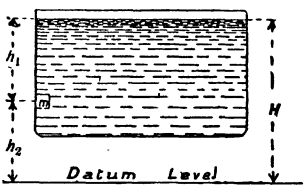
$\therefore$ **Net pressure on gate** $= 2{,}625$ "

The centre of pressure on the upper side is one-third of 5 feet, or 1 foot 8 inches, up from the bottom, and on the lower side, a third of 2 feet, or 8 inches. To find the resultant centre of pressure take moments about the bottom of the gate. Then, if this centre be distant x inches from the bottom :—

$$2{,}625 \times x = 3{,}125 \times 20 - 500 \times 8 = 62{,}500 - 4{,}000.$$

$$\therefore \qquad x = \frac{58{,}500}{2{,}625} = 22{\cdot}3 \text{ ins.} = 1 \text{ ft. } 10{\cdot}3 \text{ ins.}$$

Energy of Still Water.—When water is at rest it possesses potential energy in virtue of its position and of its pressure. Consider a tank filled with water, and imagine a small mass m of the water to escape from the tank. This mass will not only lose potential energy through falling to a lower level, but it could also do work because of the pressure of the rest of the water pushing it away.



Energy of Still Water.

It is convenient to assume some datum level at which we take the energy of position as zero.

Let H = Height of free surface above the datum level.
" h_1 = Height of free surface above m.
" h_2 = Height of m above datum level.
" g = Acceleration due to gravity.
" ρ = Density of fluid = mass of unit volume.
And, w = Weight of unit volume of fluid = ρg.

Then the work done in forcing out the mass m is :—

Energy of Pressure = *Volume* $\times$ *Pressure.*

" " $= \dfrac{m}{\rho} \times w h_1 = m g h_1.$

And, *Energy of Position* $= m g h_2$.

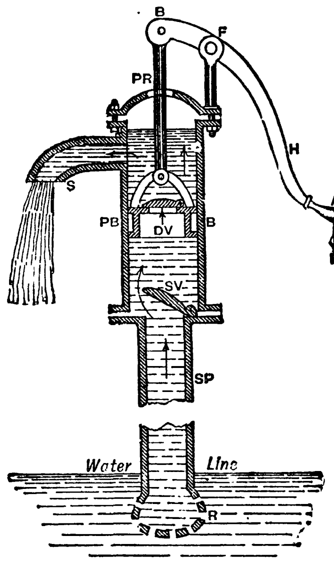
$\therefore$ *Total Energy* $= m g (h_1 + h_2) = m g \text{H}.$

Or, for a unit mass :—

Energy per unit mass $= g(h_1 + h_2) = g\,\text{H}$. (XI)

This is constant for all parts of a homogeneous fluid at rest, and H may be called the *total head* of the water in the tank.

Common Suction Pump.—This consists of a bored cast-iron barrel P B, terminating in a suction pipe, S P, fitted with a perforated end or rose R, which dips into the well from which the water is to be drawn. The object of the rose is to prevent leaves or other matter getting into the pump, that might clog and spoil the action of the valves. At the junction between the barrel and suction pipe there is fitted a suction valve S V, of the hinged clack type faced with leather. The piston or bucket B is worked up and down in the barrel of the pump by a force P, applied to the end of the handle H. This force is communicated to it through the connecting link of the hinged piston-rod, P R. In the centre and at the top of the bucket is fixed the clack delivery valve D V, which is also faced with leather in order to make it water-tight. The bucket is sometimes packed with leather; but, in the present instance, a coil of tightly woven flax rope is wrapped round the packing groove.



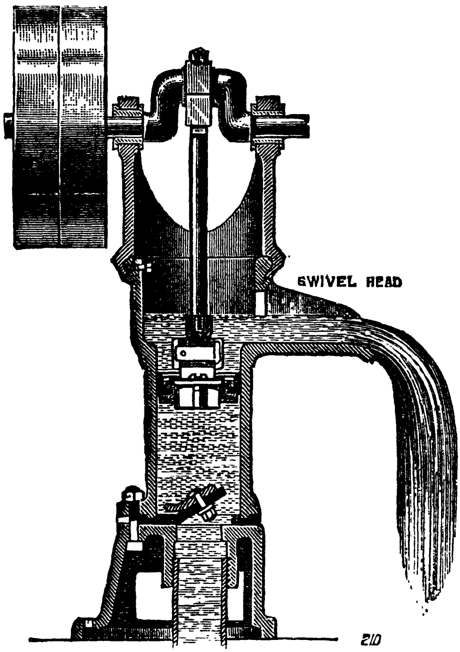
Common Suction Pump.

Action of the Suction Pump.—(1) Let the barrel and the suction pipe be filled with air down to the water-line, and let the bucket be at the end of the down stroke. Now raise the bucket to the end of the up stroke by depressing the pump handle. This tends to create a vacuum below the delivery valve; therefore, the air which filled the suction pipe opens the suction valve, expands, and fills the whole volume of the barrel. Consequently, according to Boyle's law, its pressure must be diminished in the *inverse ratio* to the enlargement of its volume. This enables the pressure of the atmosphere to force a certain quantity of water up the suction pipe, until the weight of this column of water and the pressure of the air between the suction and delivery valve, balance the pressure of the outside atmosphere.

(2) In pressing the bucket to the bottom of the barrel by ele-

vating the handle, the suction valve closes and the delivery valve opens, thereby permitting the compressed air in the barrel to escape through the delivery valve into the atmosphere.

(3) Raise and depress the piston several times so as to produce the above actions over again, and thus gradually diminish the volume of the air in the pump to a minimum. Then water will have been forced by the pressure of the atmosphere up the suction pipe and into the pump, if the bucket and the valves are tight, and if the delivery valve when at the top of its stroke be not more than the height of the hydro-barometric column above the water line of the well.*

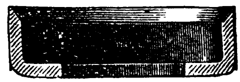


BELT-DRIVEN SUCTION PUMP.

(4) The bucket now works in water instead of in air; in fact, the machine passes from being an air-pump to being a water one. During the down stroke of the piston water is forced through the delivery valve, and during its up stroke, this water is ejected through the spout; at the same time, more water is forced up through the suction pipe and valve to fill the vacuum created by the receding piston. In the case of a common suction pump water is therefore discharged *only* during the up stroke of its piston.

* Theoretically, such a pump should be able to lift water from a depth of 34 feet below the highest part of the stroke of the delivery valve, but practically, owing to the imperfectly air-tight fitting of the piston and the valves, it is not used for withdrawing water from wells more than 20 to 25 feet below this position of the delivery valve. In fact such a pump frequently requires a bucket or two of water to be poured into it above the delivery valve in order to make it work at all, if it should have been left standing for some time without being worked.

Belt-driven Suction Pump.—When we have to raise a considerable quantity of water during a long time, then it becomes advisable to apply power derived from some prime mover. The foregoing figure illustrates an ordinary suction pump driven by a belt, with fast and loose pulleys, crank shaft, and connecting-rod. Here the suction valve and the piston are faced with leather or india-rubber, whilst the bucket valve is made of brass and ground to fit its seat. The upper portion of the pump is fitted with a swivel head, so that the pulleys may be placed fair in line with the driving pulleys.



LEATHER PACKING FOR PISTON OF SUCTION PUMP.

EXAMPLE VI.—Given two simple bucket pumps, each having a stroke of 1 foot, and cross area of bucket, 20 square inches. Suppose everything perfectly air tight, and the supply pipe 20 square inches area in one case, and 10 square inches area in the other. Neglecting friction, you are to compare the tensions on the pump rods at ends of first up stroke in each case, supposing the bucket to be 24 feet above free surface of the water in the well when at the bottom of its stroke. The supply pipe reaches to the under surface of bucket when the latter is at the bottom of its stroke.

ANSWER.—

Let p = Pressure in lbs. per *sq. ft.* on *under* surface of bucket at end of first up stroke.

„ a = Area of bucket $= \frac{20}{144}$ sq. ft.

„ h = Height of water barometer = 34 ft.

„ w = Weight of 1 cub. ft. of water $= 62\frac{1}{2}$ lbs.

„ x = Height water rises in suction pipe for first up stroke.

FIRST CASE.—*Lifting or suction pipe having an area equal to that of the bucket.*

Then at end of first up stroke, we get:—

Pressure of air on upper surface of bucket, ... } = *Atmospheric Pressure*

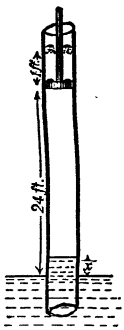
„ „ $= a\,h\,w$ *lbs.* . . . . . (1)

Pressure on under surface of bucket, $= p\;a = \left\{\begin{array}{l}\textit{Atmos. pressure}\\ -\textit{pressure of}\\ \textit{small column, } x,\\ \textit{of water}\end{array}\right.$

„ „ $= a\,(h - x)\,w.$

And, $p = (h - x)\,w.$ . . . . . (2)

Since there is exactly the same qnantity of air between the bucket and the surface of the water in the pipe at the end of the stroke as there was before the stroke commenced, we may apply Boyle's Law to determine the *volume* of air under the bucket at end of stroke.



FIRST CASE.

$$\therefore \qquad p\,a \times (25 - x) = a\,h\,w \times 24.$$

Substituting, $p = (h - x)\,w$, from equation (2), we get:—

$$(h - x)(25 - x) = 24\,h.$$

Since, $h = 34$ ft., we get, by substitution, and multiplication:—

$$x^2 - 59\,x + 34 = 0.$$

$$\therefore \qquad x = \frac{59 \pm 57{\cdot}83}{2} \text{ ft.}$$

The minus sign in the numerator of the fraction on the right-hand side of this equation is the only one admissible.

$$\therefore \qquad x = \frac{1{\cdot}17}{2} = {\cdot}58 \text{ ft. } or = 7 \text{ inches, nearly.}$$

Hence,

$$\text{Tension in pump rod} = \left\{ \begin{array}{l} \textit{Press. on upper surface of bucket} \\ \quad - \textit{Press. on under surface.} \end{array} \right.$$

$$\text{,,} \qquad \text{,,} \qquad = a\,h\,w - a\,(h - x)w = a\,x\,w.$$

$$\text{,,} \qquad \text{,,} \qquad = \frac{20}{144} \times {\cdot}58 \times 62{\cdot}5 = 5{\cdot}04 \text{ lbs.}$$

SECOND CASE.—*Lifting or suction pipe having an area equal to half that of the bucket.*

The symbols denoting the same quantities as before, we get:—

Pressure of air on upper surface of bucket $= a\,h\,w$ lbs. . . (3)

Pressure on under surface of bucket $= p\,a$

,, ,, $= a\,(h - x)\,w$. . . (4)

Vol. of air between bucket and surface of water at beginning of stroke . . $= \frac{1}{2}\,a \times 24$

,, ,, $= 12\,a$ *cub. ft.*

Vol. of air between bucket and surface of water at end of up stroke . . . $= \frac{1}{2}\,a \times (24 - x) + a \times 1$

,, ,, $= \frac{1}{2}\,(26 - x)\,a$ *cub. ft.*

∴ By Boyle's Law, we get:—

$$(h - x) w \times \tfrac{1}{2} (26 - x) a = h w \times 12 a.$$

Substituting $h = 34$, and simplifying, we get:—

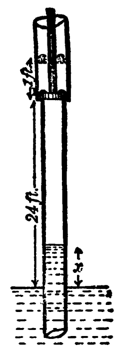
$$x^2 - 60x + 68 = 0$$

$$\therefore \qquad x = 1{\cdot}15 \text{ ft. } or = 13{\cdot}8 \text{ inches, nearly.}$$

Hence,

Tension in pump rod $= a\,x\,w.$

$$\text{,, } \qquad \text{,, } \qquad = \frac{20}{144} \times 1{\cdot}15 \times 62{\cdot}5 = 10 \text{ lbs.}$$



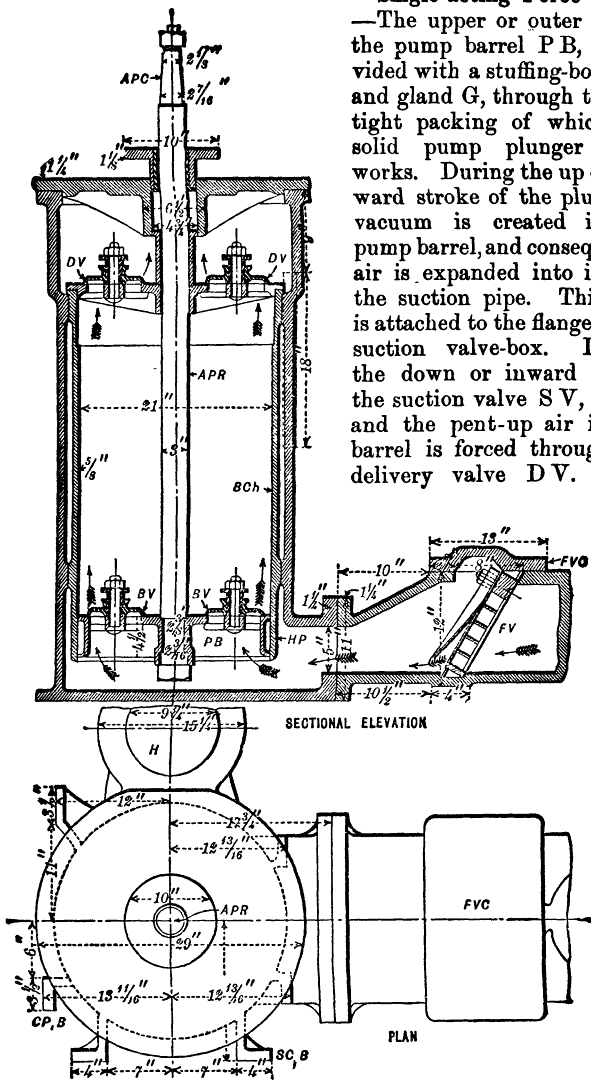
SECOND CASE.

Air Pump.—The figure on next page is a sectional elevation and outside plan of the air pump for the 1500 horse-power compound engines of the S.S. "St. Rognvald," which are fully described in the Author's *Text-Book on Steam and Steam Engines.* During the up stroke of the pump bucket P B, condensed steam and vapour are drawn from the surface condenser through the foot valve F V, into the space below the bucket, whilst any water and vapour that may have been lying above it, are forced through the delivery valves D V, into the hot well H. During the down stroke of the bucket, the water and vapour below it pass upwards through the bucket valves B V, into the space left by the descending bucket; at the same time, the foot and delivery valves automatically close on their seats. These actions take place in succession during each up and down stroke of the air pump-rod A P R, which passes through an air-tight stuffing box, and is linked to the piston-rod crosshead of the high-pressure cylinder by short connecting-rods and side levers.

The object of placing the delivery valves on the top of the air-pump barrel in addition to the ordinary bucket valves, is to cause a vacuum to be produced above the latter during the down stroke of the bucket, and thus facilitate their opening, as well as to give the vapour from the condenser a free space between these two sets of valves into which it can expand.

The cast-iron barrel of the air pump is lined with a truly bored brass chamber B Ch, the pump bucket is rendered tight by hemp rope packing H P, the foot valve is readily inspected or removed by unbolting or lifting the foot valve cover F V C, whilst the whole is bolted securely to the surface condenser bracket S C_1 B, and to the circulating pump bracket C P_1 B.

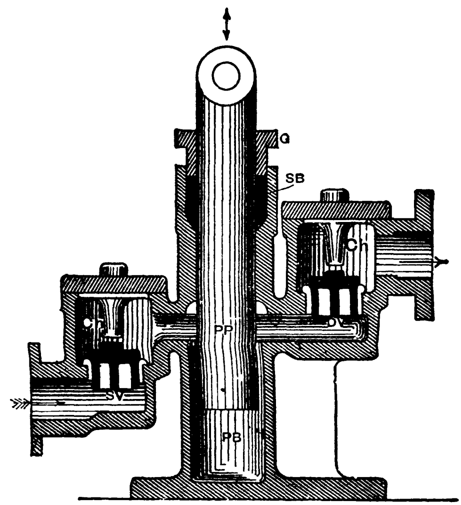
Single-acting Force Pump.—The upper or outer end of the pump barrel P B, is provided with a stuffing-box S B, and gland G, through the air-tight packing of which the solid pump plunger P P, works. During the up or outward stroke of the plunger a vacuum is created in the pump barrel, and consequently air is expanded into it from the suction pipe. This pipe is attached to the flange of the suction valve-box. During the down or inward stroke the suction valve S V, closes, and the pent-up air in the barrel is forced through the delivery valve D V. This



AIR PUMP FOR A MARINE ENGINE.

action goes on precisely in the manner just explained in the case of the suction pump, until the water rises into the barrel. Then the inward stroke of the plunger drives through the delivery valve to any desired height or against any reasonable back pressure, as in the case of a feed pump for a steam boiler.

Both the suction and the delivery valves are made of brass, and fit accurately into their brass seats. The covers to the valve chests are provided with checks Ch, to prevent the valves from rising more than the distance required to pass the water freely through them.*



SINGLE-ACTING FORCE PUMP.

The eye of the plunger may be attached to a connecting-rod actuated by a hand lever, as in the case of the common suction pump, or it may be worked from an eccentric or crank revolved by a steam engine or other motor. By whichever way it is worked, the force applied to the plunger must be sufficient to overcome the friction between the plunger and the packing, the resistance due to sucking the water from the source of supply, and of driving the same up to the place where it is delivered.

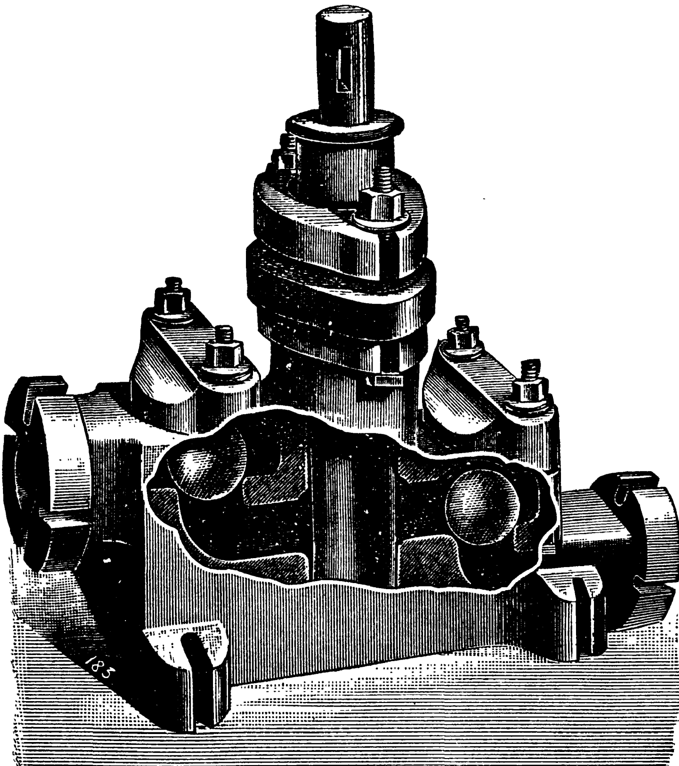
As in the case of the suction pump, the water is only delivered during each alternate stroke of the plunger, and, consequently, in an intermittent or pulsating fashion.

Very often three pumps of this kind are combined in one, each plunger being driven by a separate crank on a common shaft, and the cranks making angles of 120° with each other. Such an

* If d be the diameter of the bore of the valve seat, and h the required lift of the valve to give an opening equal in area to that bore, then h must be quarter of d. For the area of bore $= \frac{\pi d^2}{4}$ and the equivalent area of the valve opening $= \pi d h$. $\therefore \frac{\pi d^2}{4} = \pi d h$. Or, $h = \frac{d}{4}$.

arrangement is called a *three-throw pump*, and gives a very steady stream.

Single-acting Force Pump with Ball Valves.—The following illustration is a simple modification of the previous one, wherein ball valves are substituted for the common-circular three feathered type. The right-hand side forms the suction and the left-hand the delivery side. All the parts are made extra thick and strong



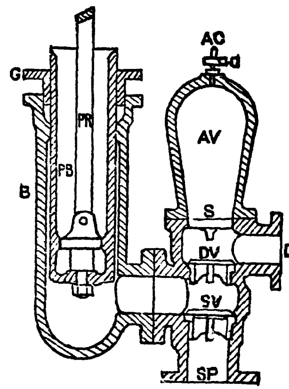
SINGLE-ACTING FORCE PUMP WITH BALL VALVES.

to resist shocks and vibrations, and most of the bolt holes have been cored to the outside of the flanges for the purpose of facilitating rapid connection and disconnection. This form of pump is much used for forcing feed-water into steam boilers, &c.

Force Pump with Air Vessel.—In the accompanying figure we

have an illustration of a force pump with both the suction and the delivery valves placed on one side of the pump barrel and then surmounted by an air vessel. The plunger, instead of being solid, as in the previous cases, is made up of a hollow trunk or barrel, with a connecting-rod fixed to an eye-bolt at its lower end.

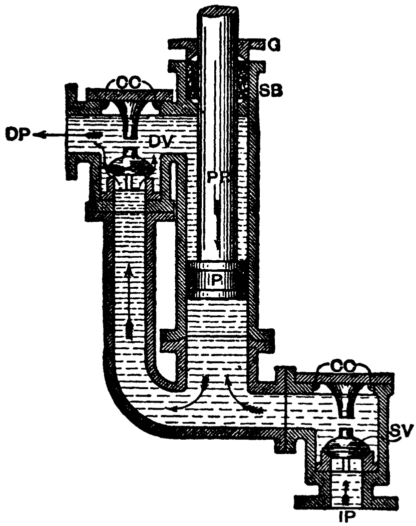
Action of the Air Vessel.—During the inward or delivery stroke of the plunger barrel P B, part of the water, which is forced from the barrel B, goes up through the delivery valve D V, into the delivery pipe D P, and the remainder enters the air vessel A V, and consequently compresses the air therein. During the outward or non-delivery stroke of the plunger the compressed air in the air vessel presses the rest of the water into the delivery pipe. In this simple way a continuous flow of water is maintained in the delivery pipe, and with far less shock, jar, and noise than in the previous cases. Where very smooth working is required, an air vessel is also put on to the suction pipe S P. Should the air in the air vessel become entirely absorbed by the water, the fact will be noticed at once, by the noise and the intermittent delivery. Then, the pump should be stopped, the air cock A C opened, and the water run out. When the air vessel is again full of air, the air cock should be shut and the pump restarted.



FORCE PUMP WITH AIR VESSEL.

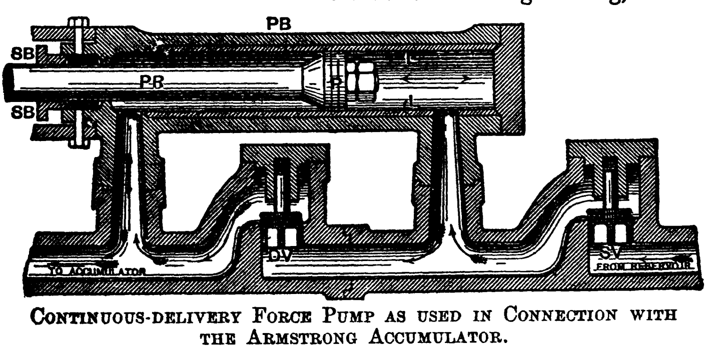
Continuous-delivery Pumps without Air Vessels.—A fairly continuous delivery of water may be obtained by making the plunger of the piston form, and the pump-rod exactly half its area; for here, during the down stroke, half the water expelled by the piston P, from the under side of the pump barrel goes up the delivery pipe D P, and the other half is lodged above the piston, to be in turn sent up the delivery pipe during the up stroke. Where very high pressures are required, such as in the filling of an accumulator ram, pumps working on this principle, but of the following form, are frequently used. The action is precisely the same as in the one just described, and the same index letters have been used, so that the student will have no difficulty in understanding the figure. The directions of motion of the piston and of the ingoing and outflowing water have been marked by straight and feathered arrows respectively.

With accumulators, and for other kinds of high-pressure **work**, it is not advisable to use air vessels, because you cannot prevent the water which enters them absorbing air and carrying the same with it to the hydraulic machines where its presence would be most objectionable. If 750 to 1000 or more lbs. pressure per square inch be generated, then you would require a very large and strong air vessel before it could be of any service. If a pressure of only 750 lbs. per square inch were used, then, since the normal pressure of the atmosphere is 15 lbs. per square inch, the air in the air vessel would be compressed, in accordance with Boyle's law, to $\frac{15}{750}$, or $\frac{1}{50}$ of its original volume. Consequently, with an air vessel of 50 cubic feet internal capacity, there would be only 1 cubic foot of air in it, when the pump was in full action.



CONTINUOUS-DELIVERY FORCE PUMP WITHOUT AN AIR VESSEL.

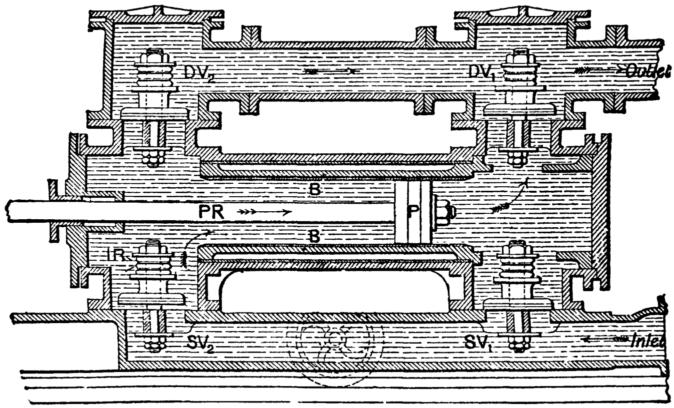
Double-acting Force Pump.—The pumps which we have hitherto considered are all single-acting, in the



CONTINUOUS-DELIVERY FORCE PUMP AS USED IN CONNECTION WITH THE ARMSTRONG ACCUMULATOR.

sense that they do not both suck and discharge water during

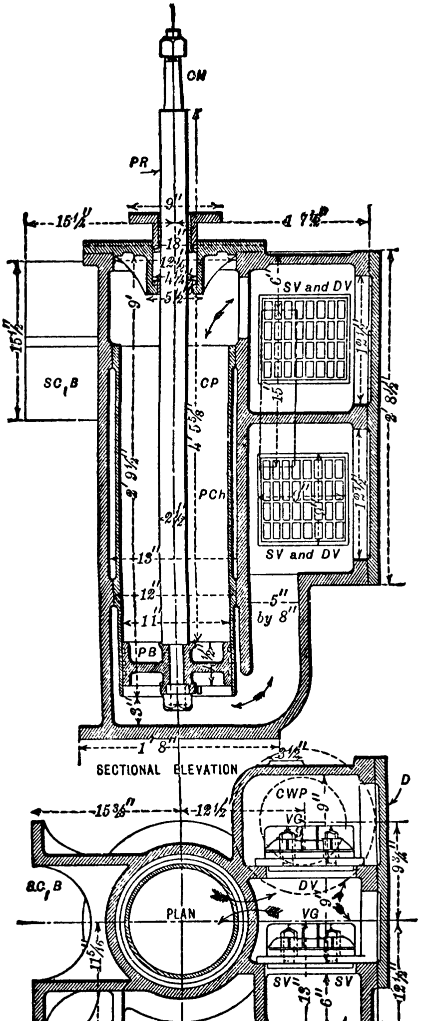
each stroke. This can, however, be accomplished by having two sets of suction and delivery valves placed at each end of the pump barrel, as shown by the accompanying figure. Here, during the outward stroke of the piston the pump draws water from the source of supply through the inlet pipe and suction valve S V_1, while, at the same time, the piston forces water in front of it through the delivery valve D V_2, and outlet pipe. During the inward stroke, suction takes place through S V_2 and discharge through D V_1, all as clearly shown by arrows in the drawing.



DOUBLE-ACTING FORCE PUMP.

The valves are provided with india-rubber cushions I R, to ease the shock and minimise the jarring noise due to their reaction and natural reverberation when they are suddenly opened and closed.

Double-acting Circulating Pump.—The following figure is a sectional elevation and plan of the circulating pump for the same marine engines as the previously described air pump. During the upstroke of the piston or pump bucket P B, water is drawn from the sea through the suction pipe S P, and the lower suction valves S V, into the lower part of the pump chamber P Ch. At the same time, the water from the top part of the chamber is forced up through the upper delivery valves D V, along the circulating water pipe C W P, into the surface condenser tubes, and from thence into the sea. During the down-stroke of the piston, the water which had previously entered by the bottom of the pump chamber is forced through the lower delivery valves D V, into the condenser tubes and sea, and at the same time, more water is taken into the



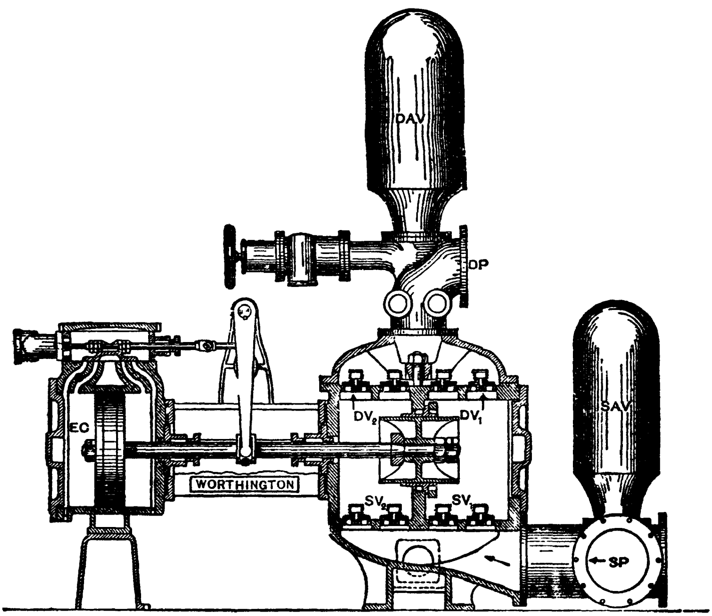
DOUBLE-ACTING CIRCULATING PUMP FOR A MARINE ENGINE.

top part of the pump chamber from the sea through the upper suction valves S V. These double actions take place in the manner described during each stroke, so that a stream of cold water is kept flowing through the condenser tubes, in order to maintain a vacuum during the exhaust of the steam from the low-pressure cylinder on to the outside of the cooled condenser tubes.

The cast-iron pump barrel of the circulating pump is lined with a truly bored brass pump chamber. The pump piston is also made of brass, and is rendered sufficiently water-tight by the simple device of turning three grooves in its outer cylindrical surface. The pump-rod passes through a water-tight gland and stuffing-box, and is connected to the reciprocating piston-rod cross-head by links and side levers, in the same way as the air pump-rod. The whole is securely bolted to the surface condenser and air pump brackets.

Worthington Steam Pump.—The perspective and sectional views

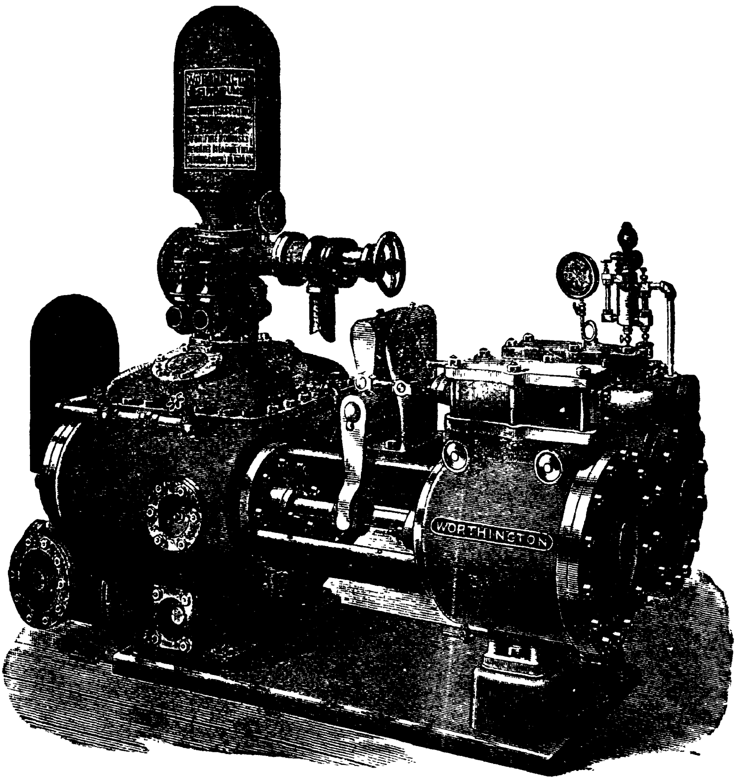
of this well-known pump, together with the following description, will serve to explain the construction and action of one of the best examples of the duplex class of steam pumps for feeding boilers, working accumulators, and hoists. It is termed a duplex pump, from the fact that it consists of two steam and two water cylinders placed side by side. The pumps draw water from the suction pipe S P, and are in this case safeguarded from shock on the suction side by an air vessel S A V. The water is admitted through the suction valves S V_1, S V_2, at each stroke respectively, and delivered by the valves D V_1, D V_2, into the discharge pipe D P, under the smoothing action of the discharge air vessel D A V.



VERTICAL SECTION OF THE WORTHINGTON STEAM PUMP.

The steam pistons and the pump plungers are directly connected together by a piston-rod, and give a swinging motion to the intermediate long levers L, which are attached by two separate spindles to two shorter levers which work the slide valve spindles. Whenever one of the steam pistons moves towards either end of its stroke the other piston is approaching the opposite end of its stroke, and by the combination of levers, piston-rods, and spindles the slide

valve of the one steam cylinder is actuated by its neighbour. The slide valves have neither lap nor lead, but immediately the piston of one cylinder covers one or other of the inner exhaust ports the steam in that cylinder is cushioned, and thus the pistons are prevented from striking their cylinder covers. Each piston as it reaches the end of its stroke automatically waits for its slide valve



PERSPECTIVE VIEW OF THE WORTHINGTON STEAM PUMP.

to be moved by the other piston-rod before it makes a return stroke. By this arrangement, the pump valves have time to close properly on their seats, and a natural smooth motion of the whole of the working parts takes place. There are no dead points in this form of duplex pump, consequently it is always ready to be started either

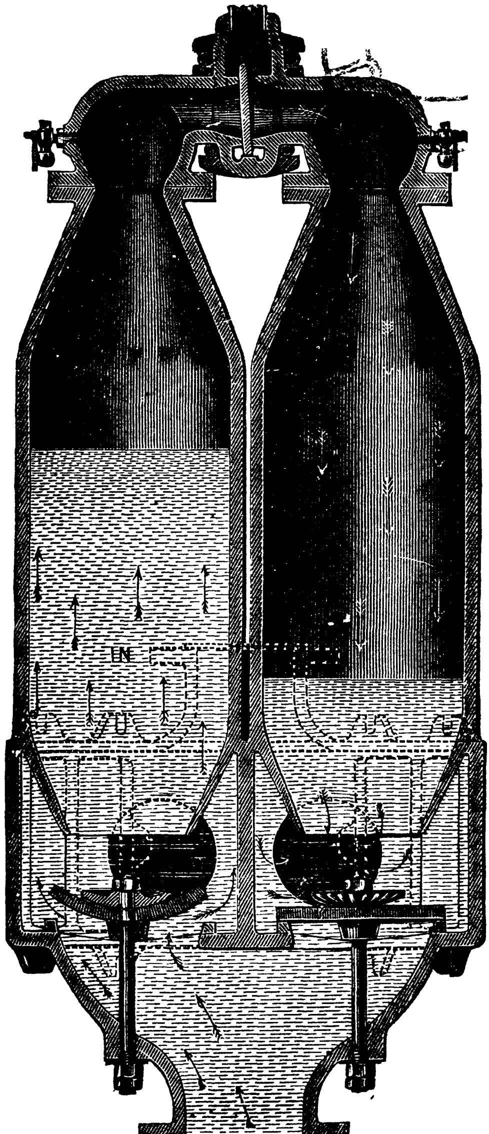
by the opening of the stop valve or by the automatic action of a float connected to the throttle valve by a chain or rope.

Pulsometer Pumps.—The very first steam pump, which was invented by Thomas Savery in 1698, had no working parts except the valves. This type has been revived for certain kinds of work in pumps of the pulsometer class, of which Bailey's "Aqua Thruster" is a good example. It consists of two long chambers, in each of which there is a valve opening upwards at the bottom, and one opening outwards at the side. At the top junction between these two chambers there is a flap valve which can put either in communication with a steam pipe while the other is shut off therefrom.

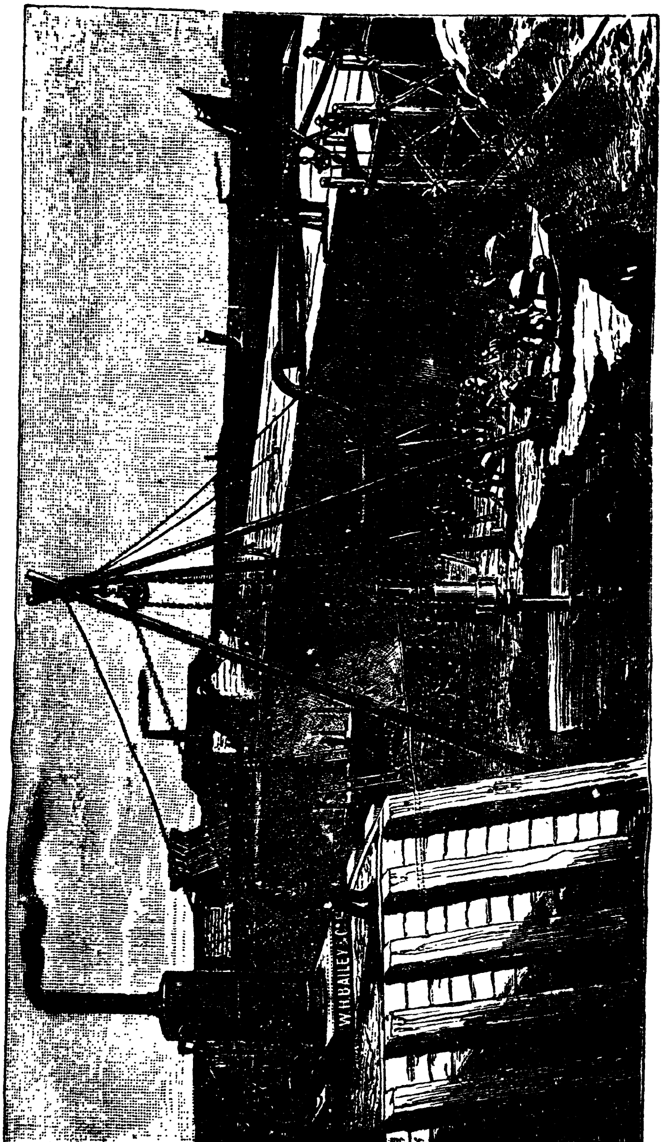
Now, suppose the right-hand chamber to be full of water while the left one is full of steam, and that the upper valve is in the position shown. Steam will enter the right-hand chamber and force the water out through the delivery valve at the side. At the same time the steam in the left compartment will be condensing, and water will therefore rise into it through the bottom valve, provided the apparatus be not too far above the free surface of the water. The inertia of this water will cause it to continue in motion after all the steam is condensed, and it will therefore compress the air that remains to a sufficient extent to shift over the valve to the other side.* If there is no air, then the water itself will strike the valve and knock it over to the other side. The conditions of the chambers are now interchanged. Water will be forced out from the left one, and fresh water will rise into the other, and the process begins again.

A large loss occurs in this kind of pump through the condensation of steam during the down stroke of the water, and also owing to the fact that the steam is used non-expansively. To reduce the former loss little cocks open into the top of the chambers and admit a little air during the time there is a vacuum inside. This air prevents the steam from coming so quickly into contact with the water as it otherwise would do, and thus reduces the loss during admission. A slight escape of steam takes place

* This is not the common explanation of the working of the pulsometer valve. It is—"As soon as the water is lowered below the upper surface of the delivery valve, steam blows through with some violence and causes a commotion and a rapid condensation in the chamber. The valve is then *drawn* to the right-hand side." This is quite wrong. The valve can only be shifted by being *pushed*, owing to the pressure on the closed side becoming greater than that on the other side, and it is difficult to see how the pressure in a chamber in direct communication with the boiler can become less than that in one where the steam is already all condensed, and where the pressure is considerably below that of the atmosphere. Besides, it is probable that in steady working the water never gets as low as the delivery valves.



PULSOMETER PUMP BY W. H. BAILEY & Co, LTD., MANCHESTER.

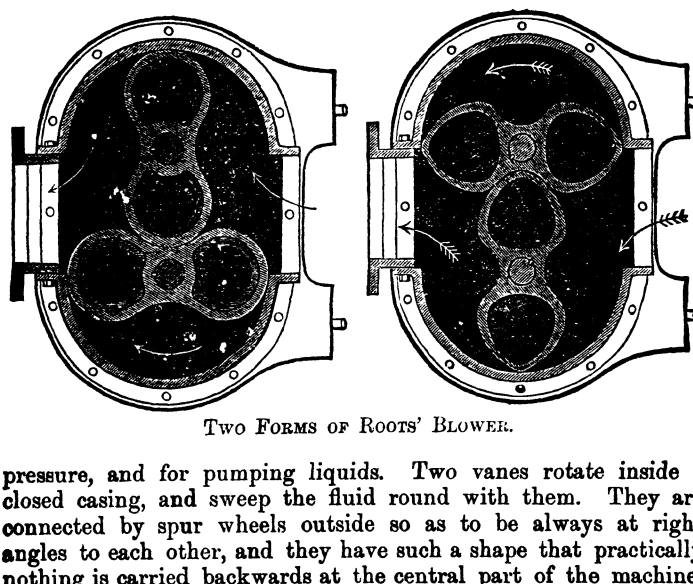


THE "AQUA THRUSTER" IN OPERATION.

through these cocks, as they are always kept open; but as their bore is so very small, the loss is less than the gain. To diminish the latter loss an extra self-acting valve, called the "grel," has been added to some forms of pulsometer with the intention of cutting off the steam earlier, and then using it expansively.

Pumps of this class are exceedingly handy for dealing with dirty water and for temporary purposes owing to their simplicity, few working parts, and the ease with which they can be erected. It is sufficient to suspend them by a chain and connect them by a pipe to a portable boiler. A suction pipe projects down below the water surface, and a flexible hose pipe will carry off the discharged water. The full page illustration shows the "Aqua Thruster" in use for pumping water from a dock during its construction.

Roots' Blower.—A form of rotary pressure pump, known as Roots' Blower, is used for obtaining a blast of air at a moderate



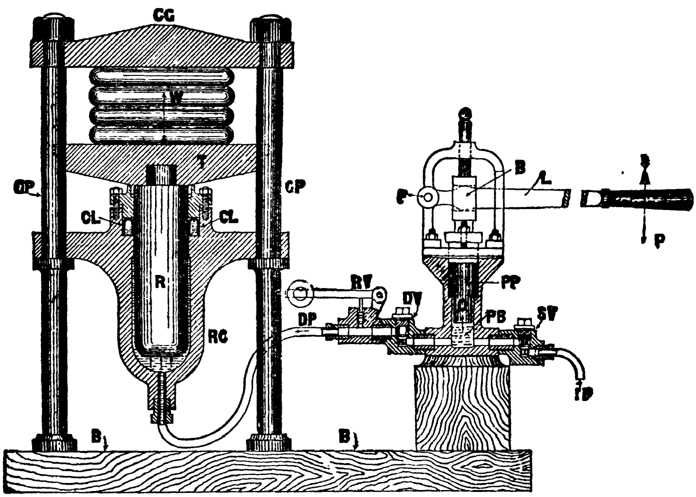
TWO FORMS OF ROOTS' BLOWER.

pressure, and for pumping liquids. Two vanes rotate inside a closed casing, and sweep the fluid round with them. They are connected by spur wheels outside so as to be always at right angles to each other, and they have such a shape that practically nothing is carried backwards at the central part of the machine. They can produce a higher pressure than an ordinary blowing fan, and are handier for many purposes than a blower of the cylinder and piston type. The student should note that this is not a centrifugal pump or fan, although there are no reciprocating parts, but simply a rotary form of pressure pump.

If a fluid be forced through this machine, then it will cause the vanes or teeth to rotate; hence it will work as a motor, and there-

fore it is a reversible machine. Many ingenious attempts have been made to produce economical steam engines on this principle; but largely owing to the difficulty of keeping them tight, they have not been so successful as their sanguine inventors expected.

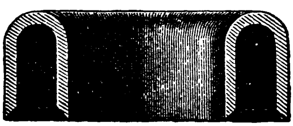
Bramah's Hydraulic Press.—This useful machine was invented by Pascal, but he could not make the moving parts water-tight. Bramah, about the year 1796, discovered a means by which this difficulty was effectually overcome; and thus the instrument has been handed down to us under his name. As may be seen from the following figure, it consists of a single-acting force pump in connection with a strong cylinder containing a plunger or ram, which is forced outwards from the cylinder through a tight collar by the pressure of the water delivered into the cylinder from the force pump.



VERTICAL SECTION OF A SMALL BRAMAH HYDRAULIC PRESS.

After what has been written about force pumps, we need not particularise about this part of the machine, except to say that the suction and delivery valve boxes at S V and D V can be disconnected from the pump, and the valve cover-checks removed at any time for the purpose of examining the parts, or of regrinding the valves into their seats. The pump plunger P P, extends through a stuffing-box and gland filled with hemp packing, and is guided by a centrally bored bracket bolted to the top flange of the

pump. The lever L, fits through a slot in this guide-bar, whereby it has an easy free motion, when communicating the force applied through it to the pump plunger. The relief valve R V, has a loaded lever so adjusted as to rise and let the water escape when the pressure exceeds a certain amount. It may also be used for ascertaining the pressure on the object under compression, or for lowering the ram R, by simply lifting the little lever and pressing down the table T, when the water flows easily from the ram cylinder R C, by the delivery pipe D P, and the relief valve. The delivery pipe is made of solid drawn brass, and the ram cylinder is carefully rounded at the bottom end, instead of being flat, in order that it may be of the strongest shape.* The guide pillars G P, are securely bolted to the base B, and to the top cross girder C G, by nuts and washers.



CROSS SECTION OF ORDINARY LEATHER PACKING.

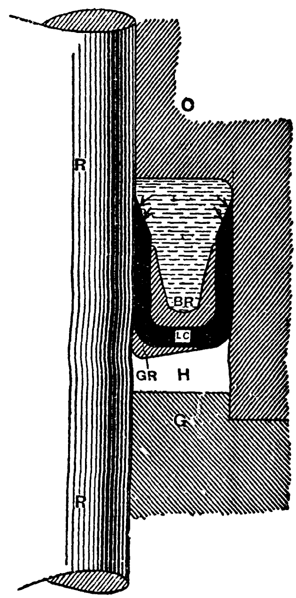
The cup leather packing C L, deserves special attention, because it formed the chief improvement by Bramah on Pascal's press. It consists of a leather collar of ∩ section, placed into a cavity turned out of the neck of the cylinder, and kept there by the gland of the cylinder cover.

This collar is made from a flat piece of new strong well-tanned leather, thoroughly soaked in water, and forced into a metal mould of the requisite size and shape to give it the form of a U collar. The central or disc portion of the leather is then cut out, and the circular edges are trimmed up to a sharp bevel as shown.

The following figure shows an enlarged section of Bramah's packing suitable for a huge press, where the desired shape of the leather collar L C, is maintained by an internal brass ring B R, and an outside metal guard ring G R, resting on a bedding of hemp H. It will be observed at once, from an inspection of this figure, that the water which leaks past the easy fit between the plunger or ram R, and the cylinder C, presses one of the sharp

* In the case of large cylinders for very great pressures, the lower or inner end of the cylinder should be carefully rounded off, both inside and outside. For, if left square, or nearly square, the crystals formed in the casting of the metal naturally arrange themselves whilst cooling in such a manner as to leave an initial stress, and consequent weakness, inviting fracture along the lines joining the inside to the outside corners of the cylinder end. The severe shocks and stresses to which this weak line of division is subjected during the working of the press would sooner or later force out the end of the cylinder, in the shape of the frustum of a cone, unless the cylinder had been made unnecessarily thick and heavy at the bottom end.

edges of the leather collar against the ram, and the other edge against the side of the bored cavity in the neck of the cylinder, with a force directly proportional to the pressure of the water in the cylinder. By this simple automatic action, the greater the pressure in the cylinder the tighter does the leather collar grip the ram and bear on the cylinder's neck.



LEATHER COLLAR FOR A LARGE HYDRAULIC PRESS.

Referring again to the figure of the Bramah press, by taking moments about the fulcrum at F, we obtain the pressure Q, on the plunger of the force pump. Neglecting weight of lever and friction, we get :—

$$P \times AF = Q \times BF.$$

$$\therefore \qquad Q = \frac{P \times AF}{BF}.$$

Further, we know that the statical pressure Q, is transmitted with undiminished force to every corresponding area of the cross section of the ram. Hence,

Q : W : : area of plunger : area of ram.

$\therefore$ W $\times$ area of plunger = Q $\times$ area of ram.

Or, $$W \times \pi r^2 = Q \times \pi R^2.$$

Where r = radius of plunger, and R = radius of ram, both in the same unit. Substituting the previous value for Q, and dividing each side of the equation by π, we get :—

$$W \times r^2 = \frac{P \times AF}{BF} \times R^2.$$

$$\therefore \qquad W = \frac{P \times AF}{BF} \times \frac{R^2}{r^2} = \frac{P \times AF}{BF} \times \frac{D^2}{d^2}.$$

Where D and d are the diameters of the ram and plunger respectively.

EXAMPLE VII.—In a small Bramah press, P = 50 lbs., A F = 20 ins., B F = 2 ins., area of plunger = 1 sq. in., whilst area of

ram = 14 sq. ins. Find W, neglecting friction and weight of lever.

Answer.—By the above formula :—

$$W = \frac{P \times AF}{BF} \times \frac{R^2}{r^2}.$$

$$\therefore \qquad W = \frac{50 \times 20}{2} \times \frac{14}{1} = 7{,}000 \text{ lbs.}$$

Example VIII.—In Bramah's original press at South Kensington the plunger is 3 ins. in diameter, and it acts at a distance of 6 ins. from the fulcrum, which is at one end of a lever 10 ft. 3 ins. long, carrying a loaded scale-pan at the other end. What should be the pressure of the water in the press in order to lift a weight of 3 cwts. in the scale-pan, neglecting the weight of the lever? Make a diagram of the arrangement.

Answer.—Here $d = 3$ ins., consequently the area of the plunger $= \frac{\pi}{4} d^2 = \cdot 7854 \times 3 \times 3 = 7$ sq. ins., and B F = 6 ins.; A F = 10 ft. 3 ins. = 123 ins.; P = 3 cwts. = 3 × 112 = 336 lbs. Now we have to find the pressure per sq. in. on the ram that will balance P, acting with the stated advantage, since the area of the ram is not given.

By the above formula :—

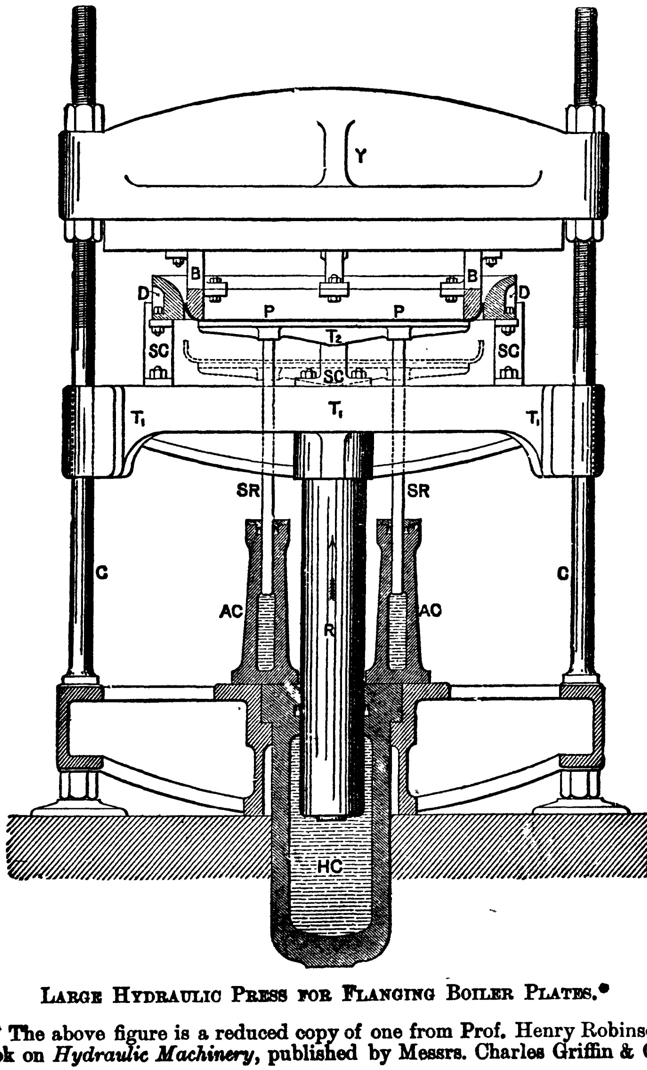
$$W = \frac{P \times AF}{BF} \times \frac{\text{area of 1 sq. in.}}{\text{area of plunger}} = \frac{336 \times 123}{6} \times \frac{1}{7}$$

Or, W = 984 lbs. per sq. in.

Hydraulic Flanging Press.—As an example of the practical application of the Bramah press to modern boiler-making, the accompanying illustration shows the form which it takes when used for flanging. It is worked by a high-pressure water supply derived from a central accumulator, which may at the same time be used to work cranes, punching, riveting, and other similar machine tools.

The operation of flanging the end tube-plates of a locomotive boiler is carried out in the following manner :—The ram R is lowered to near the bottom of the hydraulic cylinder H C, in order to leave room to place the heated boiler plate on the movable table T_2. High-pressure water is then admitted from the central accumulator to the auxiliary cylinders A C, thus forcing the side rams S R, with their table T_2, and the plate P vertically upwards,

until the upper surface of the plate bears hard against the bearers B, or internal part of the dies. Water from the same source is now admitted into the hydraulic cylinder H C, and forces up the ram R with its table T_1, supporting columns S C, and the external



LARGE HYDRAULIC PRESS FOR FLANGING BOILER PLATES.*

* The above figure is a reduced copy of one from Prof. Henry Robinson's book on *Hydraulic Machinery*, published by Messrs. Charles Griffin & Co.

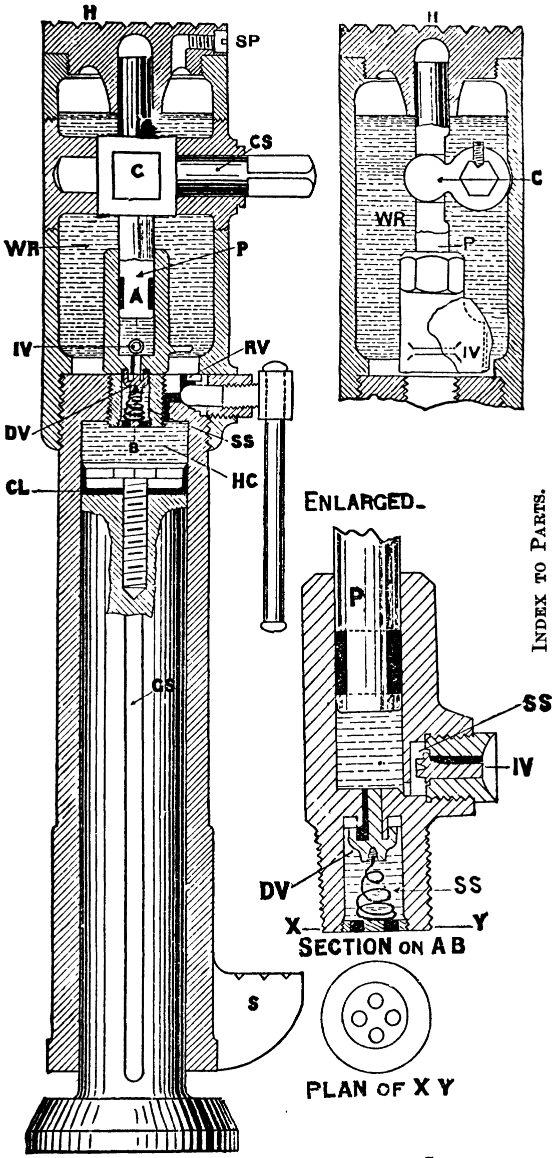
part of the disc D, until the latter has quietly and smoothly bent the heated edge of the plate round the curved corner of the internal bearer B. The ram R is now lowered, carrying with it the table T_1 and dies D, by letting out the water from H C. Then the table T_2 with the flanged plate is lowered by letting out water from A C. The plate is removed from its table, allowed to cool, faced, placed in position in the barrel of the boiler, marked off for the rivet holes, drilled, rimed, and riveted in the usual manner. The student will thus understand what a useful and powerful tool a hydraulic press is to the engineer in the hands of a skilful workman; for, it can be made to do better work in far less time, and with far greater certainty, uniformity, and exactitude, than the boiler-smith can turn out, with any number of hammermen to help him. It is fast replacing the steam-hammer for cressing work, and also steam or belt-driven punching and riveting machines, steam cranes, screw and wheel-gear hoists, as well as the screw press for making up bales of goods.

Hydraulic Jack.—This is a combined force pump and hydraulic press arranged in such a compact form as to be readily portable, and applied to lifting heavy weights through short distances. It therefore effects the same objects as the screw-jack, but with less manual effort and with greater mechanical advantage.

The base on which the jack rests is continued upwards in the form of a cylindrical plunger, so as to constitute the ram of the hydraulic cylinder H C. Along one side of this ram there is cut a grooved parallel guide slot G S, into which fits a steel set pin, screwed through the centre of a nipple cast on the side of the cylinder (not shown in the drawings) for the purpose of guiding the latter up and down without allowing it to turn round. The top of the ram has bolted to it a water-tight cup leather C L, by means of a large washer and screw-bolt.

The action of this cup leather is precisely the same as the leather collar in the cylinder of the Bramah press already described; but it has only to be pressed by the water in one direction —viz., against the sides of the truly-bored cast-steel cylinder, instead of against both the ram and the cylinder neck, as in the former case. The head H and upper portion of the machine is of square section, and is screwed on to the hydraulic cylinder in the manner shown by the figure. It contains a water reservoir W R, which may be filled or emptied through a small hole by taking out the screw plug S P.* In the centre line of the head-

* This screw plug S P is slackened back a little to let the air in or out of the top of the water reservoir when working the jack. There is generally another and separate screw plug opening for filling or emptying the water reservoir, quite independent of the above-mentioned one, which is used in this case for both purposes.



THE HYDRAULIC JACK.

INDEX TO PARTS.

H represents Head for placing under a load.		D V represents Delivery valve.	
SP ,, Screw plug for filling and emptying [W R.		S S ,, Spiral springs for I V and D V.	
W R ,, Water reservoir.		R V ,, Relieve valve.	
C ,, Crank.		H C ,, Hydraulic cylinder.	
CS ,, Crank shaft.		C L ,, Cup leather-packing.	
P ,, Pump plunger.		G S ,, Guide slot along the length of arm.	
I V ,, Inlet valve.		S ,, Step for placing under a load.	

piece there is placed a small force pump, the lower end of which is screwed into the centre of the upper end of the hydraulic cylinder. This pump is worked by the up-and-down movement of a handle placed on the squared outstanding end of the turned crank shaft C S. To the centre of the crank shaft there is fixed a crank C, which gears with a slot in the force-pump plunger P, and thus the motion of the handle is communicated to the pump plunger. By comparing the right-hand section of the water reservoir, and the section on the line A B, with the vertical left-hand section of the jack, it will be seen where the inlet and delivery valves I V and D V are situated. On raising the pump plunger P, water is drawn from W R into the lower end of the pump barrel through I V, and on depressing the plunger this water is forced through the delivery valve D V, into the hydraulic cylinder, thus causing a pressure between the upper ends of the cylinder and the ram, and thereby forcing the cylinder, with its grooved head H, and footstep S, upwards, and elevating whatever load may have been placed thereon. Both the inlet and outlet valves are of the kind known as "mitre valves." They have a chamfer cut on one or more parts of their turned spindles, so as to let the water in and out along these channels. The valves are assisted in their closing action by small spiral springs S S, bearing in small cups or hollow centres, as shown more clearly in the case of D V by the enlarged section on A B.

When it is desired to lower the jack, the relief valve R V is screwed back and the water is thus allowed to be forced up again into W R.

Example IX.—Mr. Croydon Marks, in his book on *Hydraulic Machinery*, illustrates and describes another method of lowering the jack-head (first introduced by Mr. Butters, of the Royal Arsenal, Woolwich), where, by a particular arrangement, the inlet and delivery valves are acted upon by an extra depression of the handle, and consequent movement of the pump plunger. He also gives the main dimensions, with a drawing, of the standard 4-ton pattern as used by the British Government, where the ram has a diameter $D = 2$ ins., the pump plunger a diameter $d = 1$ in.; and the ratio of the leverage of the handle to the crank is 16 to 1. Therefore, from the previous formula we find that:—

$$\textit{The Theoretical Advantage} = \frac{W}{P} = \frac{A\,F}{B\,F} \times \frac{D^2}{d^2} = \frac{16}{1} \times \frac{2^2}{1^2} = \frac{64}{1}.$$

And he instances two trials by Mr. W. Anderson, the Inspector-General of Ordnance Factories, to determine the efficiency of these jacks, where, with a pressure on the end of the working handle of

76 lbs., the theoretical load should have been 76 lbs. × theoretical advantage = 76 × 64 = 4,864 lbs., instead of which it was only 3,738 lbs. :—

$$\therefore \qquad 4{,}864 \text{ lbs.} : 3{,}738 \text{ lbs.} : 100 : x.$$

$$\text{Or,} \qquad x = \frac{3{,}738 \times 100}{4{,}864} = 77 \text{ per cent. efficiency.}$$

In a second trial, a load of 1,064 lbs. required a pressure of 22 lbs. on the handle, and consequently the efficiency at this lighter load, as might be expected, was less, or only 74 per cent.

EXAMPLE X.—With a hydraulic jack of the dimensions given above, and of 77 per cent. efficiency, it is desired to lift a load of 4 tons; what force must be applied to the lever handle?

ANSWER.—By the previous theoretical formula :—

$$W = \frac{P \times AF}{BF} \times \frac{D^2}{d^2}$$

$$\therefore \qquad P = \frac{W \times BF}{AF} \times \frac{d^2}{D^2}$$

$$,, \quad = \frac{4 \times 2{,}240 \times 1}{16} \times \frac{1^2}{2^2} = 140 \text{ lbs.}$$

But the efficiency of the machine is only 77 per cent., consequently 140 lbs. is 77 per cent. of the force required :—

$$\therefore \qquad 77 : 100 :: 140 \text{ lbs.} : x \text{ lbs.}$$

$$x = \frac{140 \times 100}{77} = 181{\cdot}8 \text{ lbs.}$$

EXAMPLE XI.—Show, with the aid of sectional sketches, the construction of the ordinary hydraulic lifting jack. If, in such a machine, the mechanical advantage of the lever or handle is 12 to 1, and the diameter of the lifting ram is 2 inches, while the diameter of the plunger is $\frac{7}{8}$ of an inch, what weight can be lifted theoretically when a pressure of 50 lbs. is applied to the lever handle?

ANSWER.—The hydraulic jack has been fully described and illustrated in this lecture.

Let D = Diameter of ram = 2 inches.
,, d = ,, plunger = $\frac{7}{8}$ inch.
,, n = Mechanical advantage of lever = 12 : 1.
,, P = Effort applied at end of lever = 50 lbs.
,, W = Weight raised.

If Q denotes the pressure on the plunger, caused by the effort P applied at the end of the lever, then :—

$$Q = P\,n.$$

But, $$\frac{W}{Q} = \frac{\text{area of ram}}{\text{area of plunger}} = \frac{D^2}{d^2}.$$

$$\therefore \quad \frac{W}{P\,n} = \frac{D^2}{d^2}.$$

This is a formula giving the relation between W, P, *n*, D, and *d*, which is also true for the hydraulic press.

Substituting the above values in this formula, we get :—

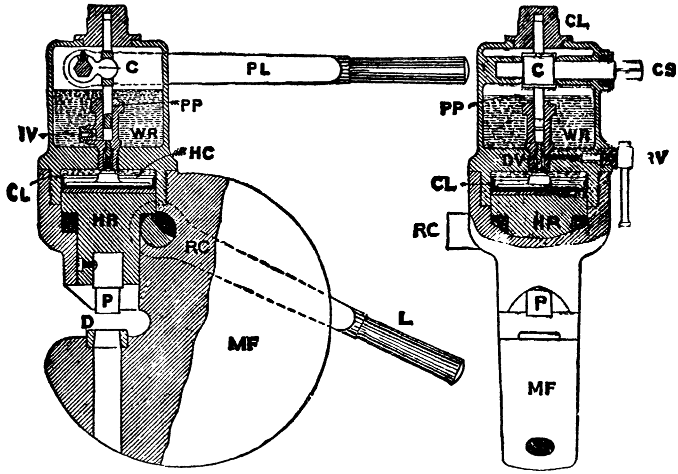
$$\frac{W}{50 \times 12} = \frac{2^2}{(\frac{7}{8})^2}.$$

$$\therefore \quad W = 3{,}134 \text{ lbs., } or = 1{\cdot}4 \text{ tons, nearly.}$$

Hydraulic Bear.—This is another very useful application of the hydraulic press and force pump. It is used in every shipbuilding-yard and bridge-building works. By comparing the drawing with the index to parts, it may be seen that its construction and action are similar to the hydraulic jack just described in full detail, and we need say nothing more than direct the student's attention to the action of the raising cam, and to the means by which the apparatus is lifted and suspended. In order to raise the punch P, for the admittance of a plate between it and the die D, the relief valve R V, must first be turned backwards, and the lever L, depressed. This causes the corner of the raising cam R C, to force the hydraulic ram H R, upwards, and the water from the hydraulic cylinder H C, back into the water reservoir W R. The relief valve R V, may now be closed and the plate adjusted in position. Then the pump lever P L, can be worked up and down until the punch P, is forced through the plate, and the punching drops through the die hole D, in the metal frame M F, to the ground, or into a pail placed beneath to receive it.

The whole bear is suspended by a chain (worked by a crane or other form of lifting tackle) attached to a shackle, whose bolt passes through a cross hole in the back of the metal frame M F, just above, but a little to the front of, the centre of gravity of the machine. This hole and shackle are not shown in the drawing, but the student can easily understand that the hole would be bored a little above where the letters R C, appear on the side view,

and that the chain would pass clear of the pump lever, since this works well to the right-hand side of the bear.



SIDE VIEW AND SECTION. END VIEW AND SECTION.

THE HYDRAULIC BEAR, OR PORTABLE PUNCHING MACHINE.

INDEX TO PARTS.

P L	represents	Pump lever.	H C	represents	Hydraulic cylinder.
C S	,,	Crank shaft.	C L	,,	Cup leather.
C	,,	Crank.	H R	,,	Hydraulic ram.
P P	,,	Pump plunger.	R C	,,	Raising cam.
W R	,,	Water reservoir.	L	,,	Lever for R C.
I V	,,	Inlet valve.	P	,,	Punch.
D V	,,	Delivery valve.	D	,,	Die ring.
R V	,,	Relief valve.	M F	,,	Metal frame.

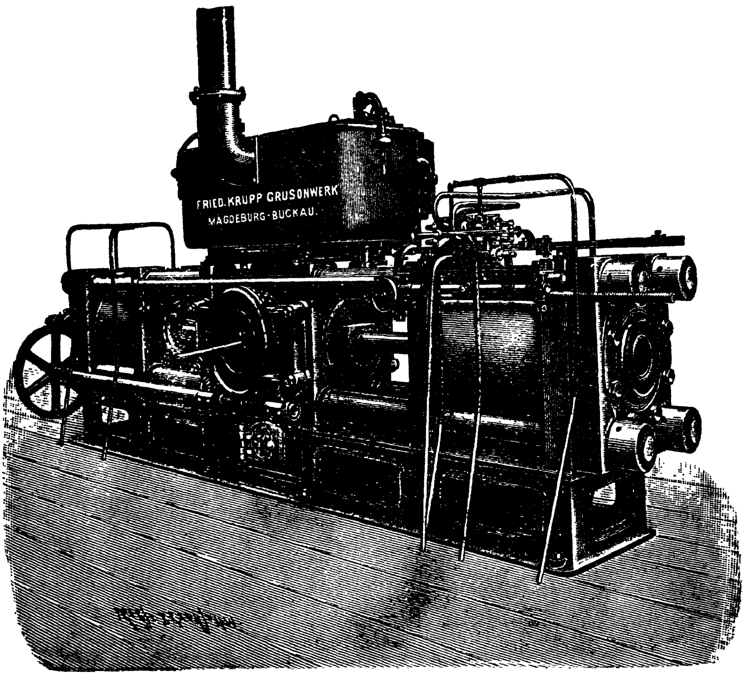
Lead-covering Cable Press.—Hydraulic presses are now employed for making lead pipes, and for covering electric light cables with a close-fitting tube of lead. For the former purpose the lead is heated until it is nearly melted in a strong chamber from which it is forced by rams to squirt through a die at the top of the machine. A mandrel projects into the centre of the die, and, consequently, the lead issues as a continuous tube.

The accompanying figure illustrates a press for covering electric cables with lead in this manner, as carried out by Messrs. Siemens Brothers, at their Woolwich works. It consists essentially of a receiver in which two rams work, and which contains a mandrel

and matrice. The lead is melted in the melting pot at the top by gas or petroleum, and is then poured into the receiver below.

The cable enters at one side of the receiver, and passes through the mandrel and leaves at the other, while as soon as the rams begin to force their way into the receiver, the lead casing is formed round the cable.

The matrice can be so nicely adjusted by means of steel cones, that a lead casing of perfectly uniform thickness of a fraction of a millimetre, can be obtained. An excellent feature of this press is,



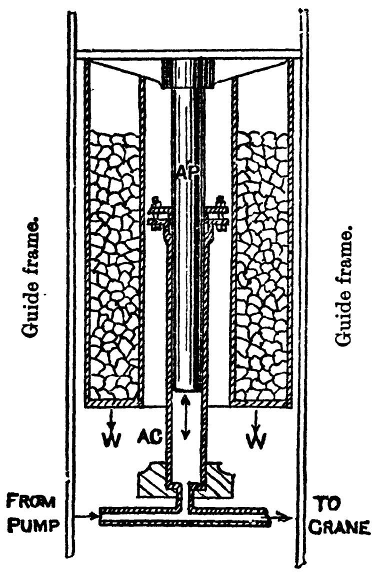
PRESS FOR COVERING CABLES WITH LEAD, BY FRIED. KRUPP GRUSONWERK, GERMANY.

that it possesses two rams which, by distributing the pressure more uniformly, ensure a more regular casing than is possible with only one, as is usually the case. The two rams are connected together by an adjusting apparatus so that they will both move forward at the same rate. The mandrels and matrices can easily be changed to suit cables whose diameters range from $\frac{1}{8}$ in. to $2\frac{3}{4}$ ins.

Hydraulic Accumulator.—The demand for hydraulic power to work elevators, cranes, swing bridges, dock gates, presses, punching and riveting machines, &c., being of an intermittent nature—at one moment requiring a full water supply at the maximum pressure, and at another a medium quantity, whilst in many cases all the machines may be idle—it is evident that if an engine with pumps were devoted to supplying this demand in a direct manner, the power thereof would have to be equal to the greatest requirements of the plant, and would have to instantly answer any and every call from the same. In the case of a low-pressure supply, as for lifts, this difficulty is best overcome by placing one tank in an elevated position at the top of the hotel or building where the lift is required, and another tank below the level of the lowest flat. Then a small gas engine working a two or three-throw pump, or a Worthington duplex steam pump, may be used to elevate the water more or less continuously from the lower to the higher tank. The "head" of water in the elevated tank will, if sufficient, work the lift at the required speed, and the discharged water from the hydraulic cylinder will enter the lower tank, to be again sent round on the same cycle of operations. Should the lift be stopped for any considerable time, then a float in the upper tank, connected by a rope or chain with the shifting fork for the belt-driven pumps (in the case of the gas engine) will force the belt over on to the loose pulley, or shut off the steam from the Worthington pump. And when the water falls in the upper tank, the float will cause a reverse movement of the rope and shift the belt to the tight pulley, or open the steam valve, and so start the pumps.

When the pressures required are great, such as for cranes, &c., where 700 lbs. on the square inch is considered a very medium pressure, an elevated tank would be out of the question, for it would have to be fully 1600 feet high in order to exert this force and to overcome friction. Under these circumstances recourse is had to a very simple and compact arrangement called an accumulator, of which we here give a lecture diagram, without any details of cocks or valves, and automatic stopping and starting gear. A steam engine, or other motor, works a continuous delivery pump, of the combined piston and plunger type, without an air vessel, as already illustrated in this lecture. The water from the pump enters the left-hand branch pipe leading into the foot of the accumulator cylinder, and forces up the accumulator ram with its crosshead or top T-piece, and the attached weight or dead load, until the ram has reached nearly to the end of its stroke. Then the top of the T-piece, or a projecting bracket on the side of the wrought-iron cylinder containing the dead load, engages with and lifts a small weight attached to a chain passing over a pulley

fixed to the guide frame or to the wall of the accumulator house. This chain is connected directly to the throttle valve of the steam supply pipe, or to the belt-shifting gear if the pump is driven by belt gearing, and being provided with a counter-weight, the motor and pump are automatically stopped by the raising of the weight and the chain in the accumulator house. Should the water which has been forced into the accumulator cylinder be now used by a crane or other machine, the load on the ram causes it to follow up and keep a constant pressure on the water. The starting weight falls as the receding T-piece or bracket descends, and thus pulls the starting chain, and opens the steam engine throttle valve, or shifts the belt from the loose to the fixed pulley, and again sets the pump to work. Should the hydraulic machines be working continuously, then the pump is kept going, for the water from it passes directly on to the machines, and only the surplus water finds its way into the accumulator cylinder if the pump's supply exceeds the demand of the machines for water.



THE HYDRAULIC ACCUMULATOR.

INDEX TO PARTS.

A C for Accumulator cylinder.
A P ,, Accumulator plunger or ram.
W ,, Weight or load contained in an annular cylinder of wrought iron and suspended from the top of T-piece or crosshead.

The annular cylinder of wrought iron is generally filled with scrap iron, iron slag, sand, or other inexpensive heavy material. The accumulator cylinder A C, has a stuffing-box and gland at its upper end. A coil of hemp woven into a firm rectangular section and smeared with white lead is placed in the bottom of the stuffing-box. The gland is screwed down on the top of this packing until at the normal pressure the water in the cylinder cannot leak past it. Cup leather packing is seldom used for this simple form of accumulator; just the ordinary packing that would be used for pump rods is found to answer all requirements. This is the

simplest form of accumulator which we have here described, but other forms will be illustrated in the next lecture.

Example XII.—Describe and sketch in section an hydraulic accumulator, showing how the ram is kept tight in the cylinder. An hydraulic press, having a ram 16 inches in diameter, is in connection with an accumulator which has a ram 8 inches in diameter and is loaded with 50 tons of ballast; what is the total pressure on the ram of the press?

Answer.—The first part of the question is answered by the previous figure and by the text.

By Pascal's Law the pressure *per square inch* in the accumulator is equal to the *pressure per square inch in the* hydraulic press. Consequently :—

$$\frac{\textit{Total Pressure on Press}}{\textit{Total Load on Accumulator}} = \frac{\textit{Cross Area of Press Ram}}{\textit{Cross Area of Accumulator Ram}}$$

$$\frac{P}{50} = \frac{\pi}{4} \times 16^2 \div \left(\frac{\pi}{4} \times 8^2\right) = \frac{16^2}{8^2}.$$

$$\therefore \qquad P = \frac{50 \times 16 \times 16}{8 \times 8} = 200 \text{ tons.}$$

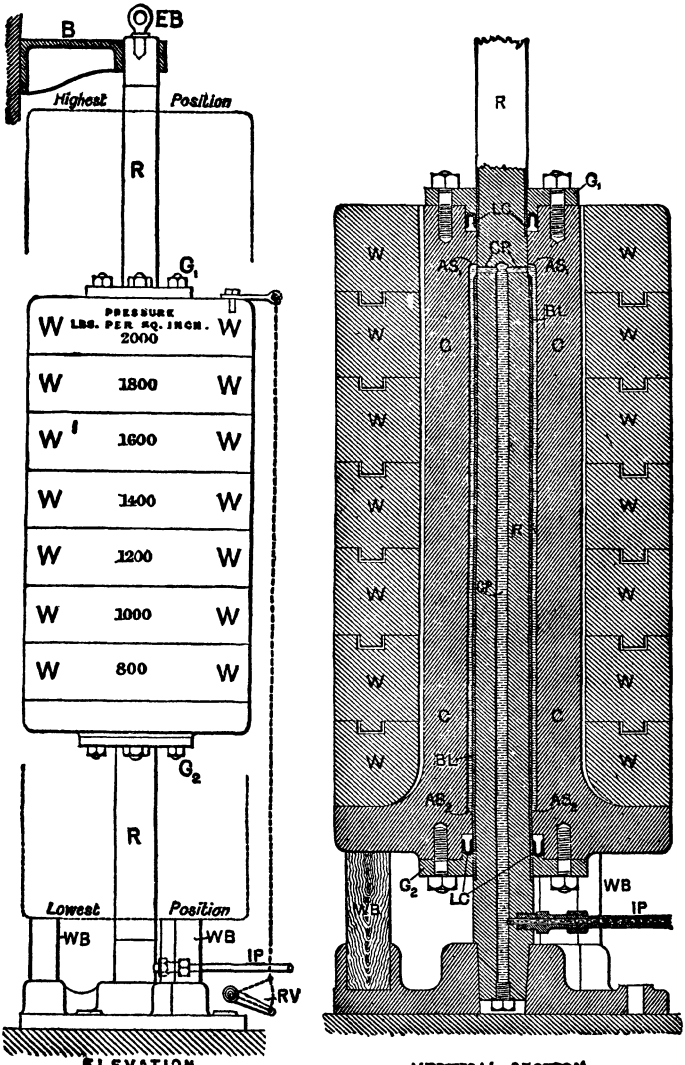
An accumulator, therefore, performs several very important functions in a most efficient manner :—

(1) It acts as a reservoir for the storage of energy.
(2) It acts as a regulator of pressure.
(3) It acts like a flywheel in taking up and giving out power in direct sympathy with the immediate wants of supply and demand.
(4) It acts as an elastic buffer, and prevents the breakage of joints, &c.
(5) It automatically controls the motive power.

The efficiency of an accumulator has been proved to be as high as 98 per cent., only 1 per cent. being lost through friction in charging, and 1 per cent. in discharging it, as tested by pressure gauges on the supply and discharge pipes. Its total store of energy is, however, comparatively small, since it is only equal to the potential energy of the weight raised. Hence, if W lbs. be the total weight raised in the accumulator, and H feet the difference of height between its highest and lowest positions, we have:—

$$\left.\begin{array}{l}\textbf{Energy Stored in}\\ \textbf{Accumulator}\end{array}\right\} = \mathbf{W\,H\ ft.\text{-}lbs.} = \frac{\mathbf{W\,H}}{\mathbf{33{,}000 \times 60}}\ \mathbf{H.P.\text{-}hours.}^{*} \qquad \text{(I)}$$

* One horse-power = 33,000 ft.-lbs. of work per minute, hence, 1 horse-power hour is 33,000 × 60 or 1,980,000 ft.-lbs.



TWEDDELL'S DIFFERENTIAL ACCUMULATOR.

EXAMPLE I.—The accumulators used in connection with the hydraulic power supply in Glasgow are 18 inches in diameter, and have a free lift of 23 feet. The total load on each is 127 tons. Find the pressure of the water and the maximum energy stored in each accumulator, neglecting friction.

ANSWER.—

$$\text{Pressure of water} = p = \frac{W}{A}.$$

$$\text{,, \quad ,,} \quad = \frac{127 \times 2{,}240}{\cdot 7854 \times 18 \times 18} = 1{,}120 \text{ lbs. per sq. in.}$$

$$\text{Energy stored} = W H = 127 \times 2{,}240 \times 23.$$

$$\text{,,} \quad = 6{,}543{,}000 \text{ ft.-lbs., or } 3\cdot 3 \text{ H.P.-hours.}$$

INDEX TO PARTS.

I P	for Inlet Pipe from pump.
R	,, Ram of accumulator.
B L	,, Brass Liner on lower part of R.
C P	,, Central and Cross Passages for water.
C	,, Cylinder.
A S_1, A S_2	,, Annular Spaces from top and bottom of cylinder.
G_1, G_2	,, Glands (top and bottom).
L C	,, Leather Cup Packings.
W	,, Weights.
W B	,, Wooden Blocks.
B	,, Bracket (at top).
E B	,, End Bearing.

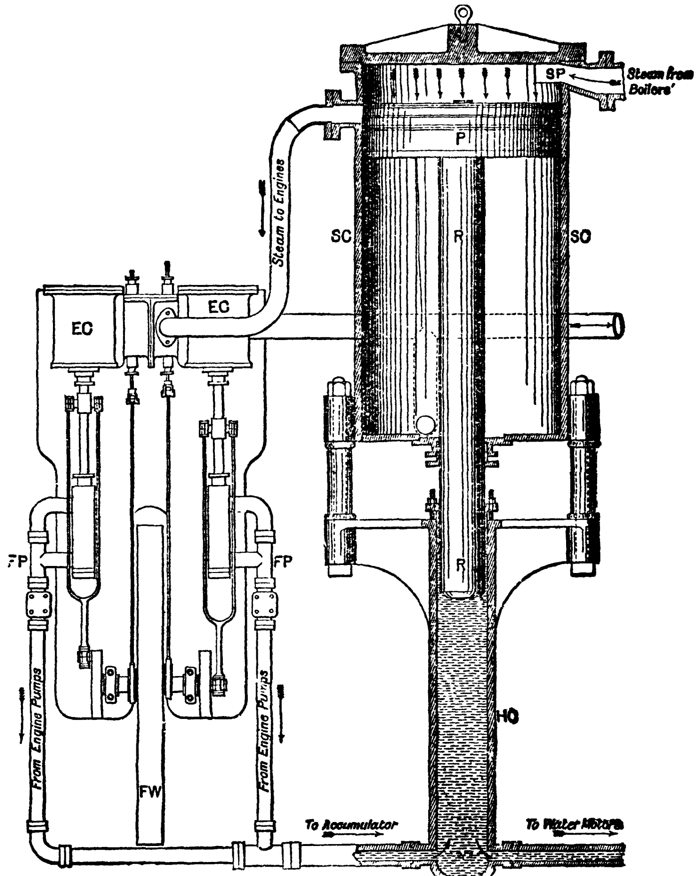
It will be seen from this example how small the total store of energy is, and that accumulators would be quite useless to maintain the supply for any length of time. One of the real advantages of the accumulator arises from the fact that we may use its energy for a very short time at a high rate. For instance, although the accumulator in the above example is only capable of maintaining 3·3 H.P. for a whole hour, it could exert 19·8 H.P. for ten minutes, or 198 H.P. for one minute.

Differential Accumulator.—Although it has only been found necessary to employ a water pressure of 400 lbs. per square inch in the charging and drawing machines for gas retorts, which we will describe later on, and, further, since it is advisable with them *to have a considerable volume of water in the accumulators* when many machines have to be worked simultaneously, yet it may not be out of place to describe here the differential accumulator designed by Mr. Tweddell for his smaller kinds of hydraulic

tools, since there may be cases in which space and compactness are of considerable importance. From the foregoing elevation and enlarged vertical section it will be seen that the ram R consists of a vertical fixed shaft secured at the top by a bracket B, and at the bottom by a footstep. The lower half of this shaft is of larger diameter than its upper half, a brass liner B L being shrunk on the former part. Moreover, this lower portion of the ram has a central passage C P drilled axially along it, with a cross passage just above the upper end of the brass liner. Through these passages water is admitted from the inlet pipe I P, which is connected directly to the force pumps. This water finds its way into an annular space A S_1, A S_2, which is the clearance between the outside of the brass liner and the inner bore of the heavy press or cylinder C. Surrounding the outside of the cylinder are placed a number of cast-iron or lead weights W which fit into each other, and form the dead load along with the weight of the cylinder. At the top and the bottom of the cylinder there are suitable glands G_1 and G_2 containing the usual leather cup packings L C. When the machine is idle the bottom flange of the cylinder rests upon wooden beams W B. It will now be readily understood that the effective area of the ram is *only* the difference between the cross areas of the brass liner B L and the upper part of the ram R, instead of the whole area of the ram as in the previous case. Hence, a very great pressure may be obtained from a small weight. For example, should the annular area representing the difference in size between the brass liner and the upper part of the iron ram be 5 square inches, and the total weight of the cylinder and its surrounding cast-iron blocks be 2,000 lbs., then, neglecting the friction at the glands, the pressure would be $2{,}000 \div 5$, or 400 lbs. per square inch. This accumulator will store up an equal amount of energy, as in the previous case, if the dead weight and height of the lift are the same since their stored energy depends directly upon $W \times H$. The volume of water contained in the accumulator will, however, be comparatively small, and hence it will fall more quickly for a certain amount of water used by the hydraulic machines which it drives. As will be seen from the left-hand figure, a relief valve R V is worked by a chain connecting an outstanding arm on the uppermost weight to the end of the lever; also any desired pressure up to 2,000 lbs. per square inch, or more, may easily be obtained from this accumulator with a comparatively small dead load and space.

Brown's Steam Accumulator.—Another very simple form of accumulator, which has proved very effective both for land purposes and on board ship, is that designed and made by Mr. A.

Betts Brown, of the Rosebank Iron Works, Edinburgh. It consists of a steam cylinder S C fitted with a piston P and a piston-rod or ram R. Steam is supplied direct to this cylinder from the boiler and presses on the piston P in opposition to the

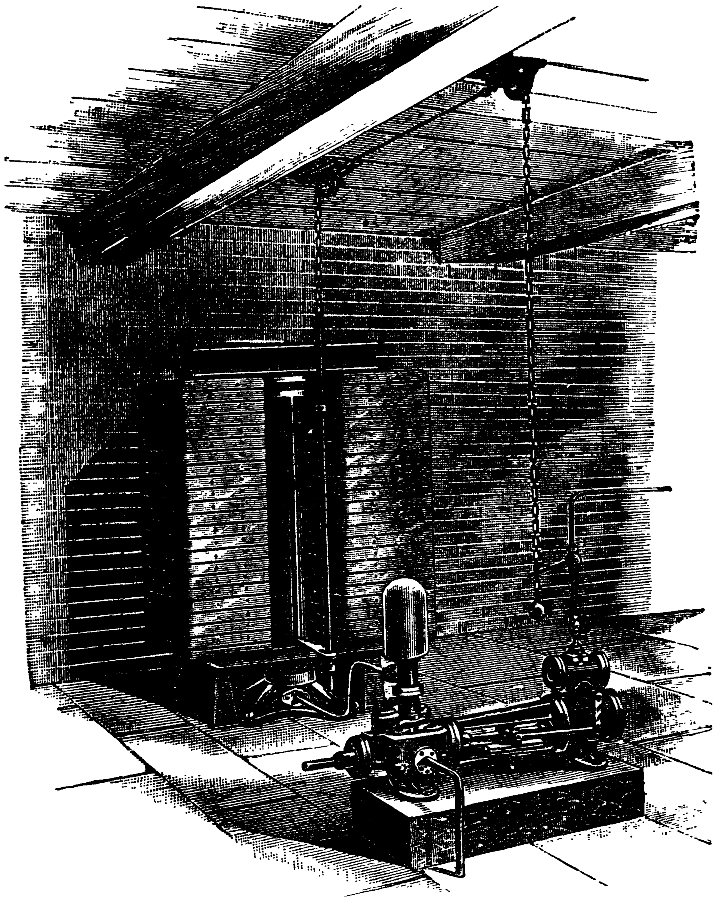


BROWN'S STEAM ACCUMULATOR OR COMPENSATED STEAM PUMP.

INDEX TO PARTS.

S P for Steam Pipe from Boilers.	H C for Hydraulic Cylinder.
S C ,, Steam Cylinder.	E C ,, Engine Cylinders.
P ,, Piston working in S C.	F P ,, Force Pumps.
R ,, Ram attached to P.	E ,, Exhaust Pipe.

water forced into the hydraulic cylinder H C by the force pumps F P, which are worked by a pair of engines. An exhaust pipe E carries away the exhaust steam from the engine cylinders E C and the bottom of the large steam accumulator cylinder S C.



SMALL HYDRAULIC ACCUMULATOR PLANT.

Suppose that the piston P is at the bottom of its cylinder, then the boiler steam not only fills the portion above the piston but passes on to the engine cylinders and therefore works the pumps

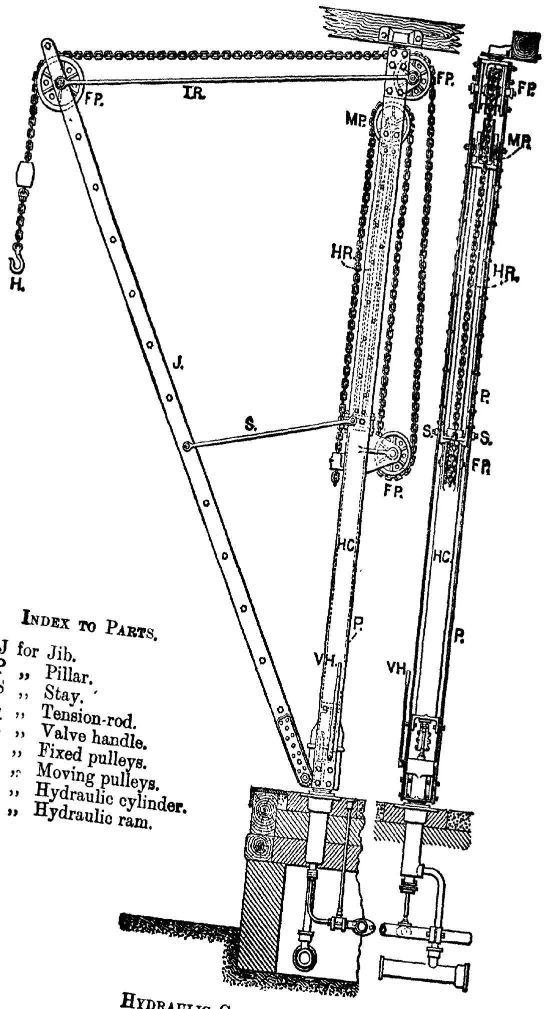
and forces up the ram and piston to the top of their stroke, when the piston gradually closes the steam passage leading to the engines until they stop. Should any of the hydraulic machines be now put into action the water flows to them from the hydraulic cylinder, the ram and piston descend and the engines are again set into motion to keep up the demand and again close the engine steam pipe. With a steam cylinder of 36 inches diameter and a ram of $9\frac{1}{4}$ inches (or, ratio of areas 15 to 1) Mr. Brown is able with about 50 lbs. steam pressure per square inch to maintain a pressure of 750 lbs. per square inch in the hydraulic mains leading to water motors, steering, stopping, and starting gear, or, in the case of a gas works, to the charging and drawing machines.

Small Hydraulic Accumulator Plant.—Should the installation need less power, a much less expensive accumulator plant may be employed, such as that illustrated. This figure is sufficiently self-descriptive after what has been said about the larger plant.

The small pumping engine, or donkey pump, which supplies water of the desired pressure to the accumulator is also shown in a perspective view. Steam is admitted to the steam cylinder by piston valves, actuated by a connecting-rod and lever, while the double-acting pump is fitted with a solid piston plunger and an air vessel.

Hydraulic Cranes.—Another very common application of hydraulic power is the working of cranes for handling goods at wharves, warehouses, or railway depôts, and we illustrate a simple one for the latter purpose. In this crane, the lower part of the pillar forms the cylinder for a ram H R, which carries a pulley M P, at its upper end and is actuated by the water pressure. A chain, fastened at one end to the crane pillar, passes over this pulley and then round two fixed ones F P, before going to the jib pulley. Hence, when the ram is forced up, the chain will be pulled up twice as far. A small slide valve, actuated by the handle V H, controls the supply of water to the hydraulic cylinder. By means of this valve, the cylinder is put into communication with either the pressure or the exhaust pipe, or it may be cut off from both. The weight of the ram and load attached to the hook H, are utilised to drive out the water on the return stroke.

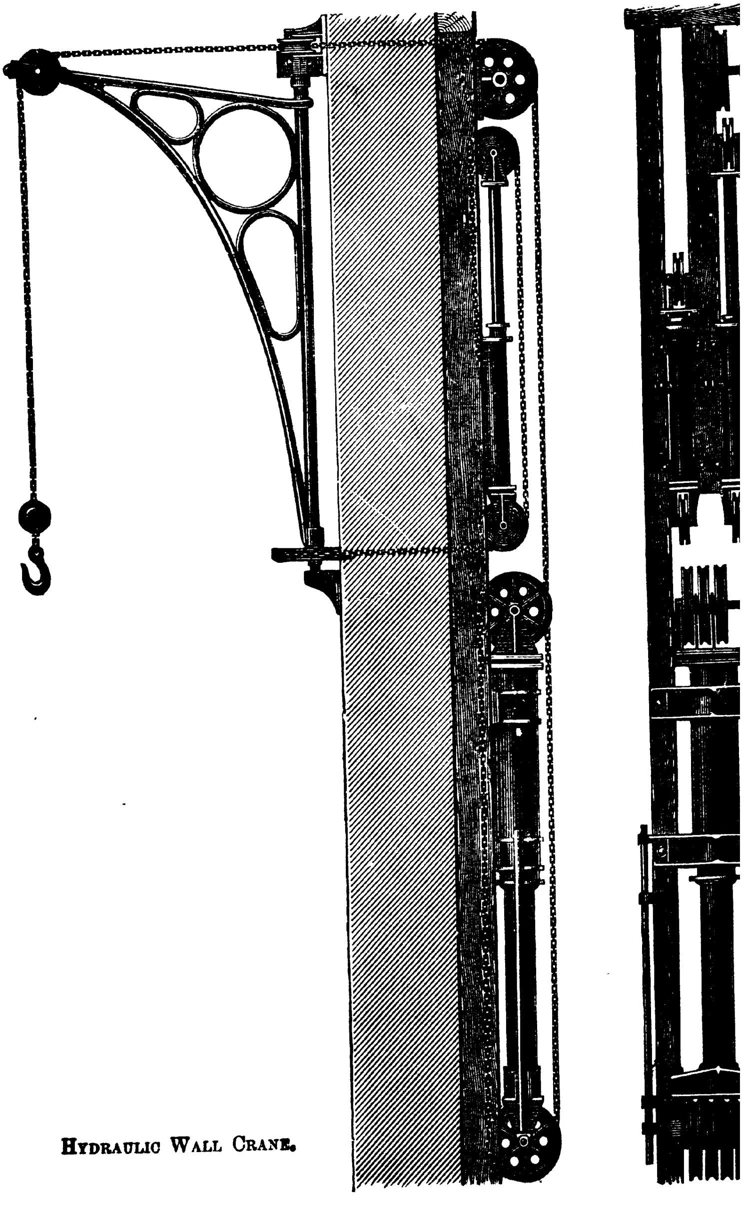
Hydraulic Wall Crane.—Our next figure shows a crane on the same principle fixed to the wall of a warehouse or shed. In this case, however, the motion of the ram is magnified eight times by placing four pulleys side by side on the end of the large ram which here moves downwards. The slewing of this crane is also accom-



Index to Parts.

J for Jib.
P ,, Pillar.
S ,, Stay.
T R ,, Tension-rod.
V H ,, Valve handle.
F P ,, Fixed pulleys.
M P ,, Moving pulleys.
H C ,, Hydraulic cylinder.
H R ,, Hydraulic ram.

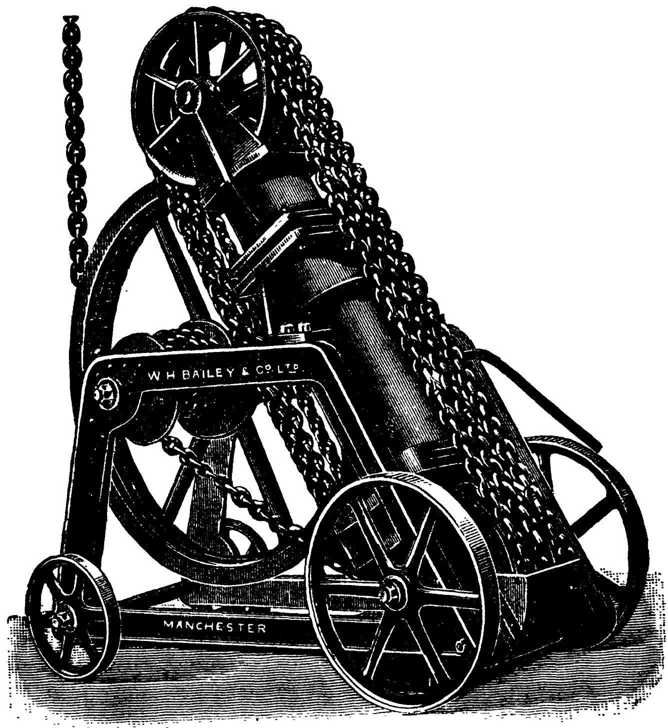
Hydraulic Crane.



HYDRAULIC WALL CRANE.

plished by hydraulic means as follows:—Each end of a chain, which passes round a wheel at the bottom of the crane, is taken over a pulley on a separate small ram, and then fixed to the framework. Matters are so arranged, that when these two smaller rams are at their half strokes the crane projects at right angles to the wall. If water be admitted below one of the rams that end of the chain is forced up and the crane is hauled round. At the same time, the other ram is pulled down by the chain so as to be ready to bring the crane back when required.

Movable Jigger Hoist.—For lifting light loads, say under a ton,



MOVABLE JIGGER HOIST BY W. H. BAILEY & CO., LTD., MANCHESTER.

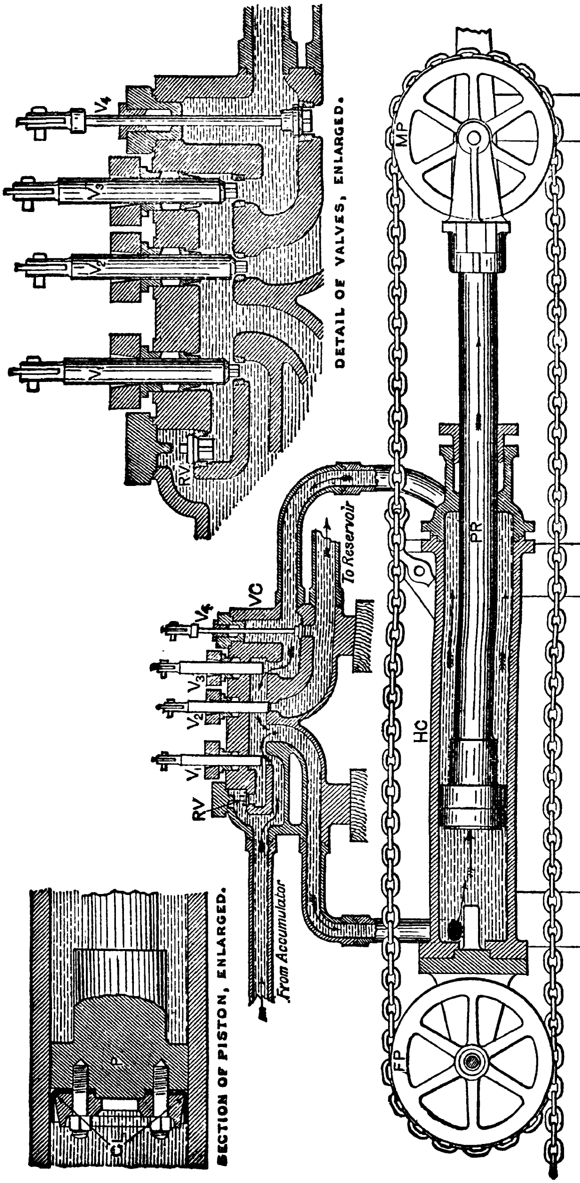
from a ship's hold, jigger hoists are often employed. The lifting chain or rope is wound on the large drum, while the ram actuates another chain coiled on a small drum on the same axis. Conse-

quently, the load is lifted very rapidly, as the motion is magnified first by the sheaves on the ends of the ram and cylinder, and then by the wheel and axle arrangement. The jigger stands on the quay and the lifting chain passes over a pulley hung from one of the ship's derricks. The valve gear can be worked, if desired, by a rope from the ship's deck.

Double Power Hydraulic Crane.—A disadvantage of hydraulic pressure apparatus is, that, as a rule, the same amount of water is used with a light load as with a heavy one, any surplus energy being consumed in fluid friction. To reduce this loss Lord Armstrong employed a combined ram and piston as shown in the figure on next page.

With the valves in the position shown, both ends of the cylinder are in communication with the supply pipe, and the ram is moved forward by the pressure on the difference between the areas of the piston and ram. The water used also corresponds to this difference of area because that in the right-hand end of the cylinder is forced round to the left, and only the remainder comes from the accumulator. Now, if we require to lift a greater load than the crane will raise under these conditions, V_3 must be closed and V_4 opened. This allows the water at the front of the piston to escape, and we get the full pressure on the back of the piston. To lower the load, V_1 is closed and V_2 opened. Then, the water escapes from the left of the piston partly to the exhaust and partly to the other end of the cylinder. The relief valve allows a little water to escape back to the supply pipe whenever the pressure in the valve chamber rises above that of the accumulator, owing to the sudden closing of the outlet valve. This gear is designed to work horizontally in a chamber sunk into the ground below the crane.

Hydraulic Capstan.—When a continuous rotary motion is required from hydraulic power, the usual method is to put three single-acting oscillating cylinders in symmetrical positions round the shaft, with all three pistons acting on one crank pin, while the motion of the cylinder itself uncovers the openings for the inflow and exit of the water at the proper time. In a common type of hydraulic capstan for use on dock quays are four cylinders with their mid positions 90° apart. A plunger in each acts on the crank below the capstan. The water enters and leaves by passages in the trunnions on which the cylinders swing, and these passages are opened at the right moment by the oscillation of the cylinder.



ARMSTRONG'S DOUBLE POWER HYDRAULIC CRANE.

H C for Hydraulic cylinder.
P R „ Piston-rod.
P „ Piston.
C L „ Cup leather.
V C „ Valve chamber.

V_1 for Inlet valve.
V_2 V_4 „ Outlet valves.
V_3 „ Connecting valve.
R V „ Relief valve.

LECTURE I.—QUESTIONS.

1. One side of a reservoir has a slope of 12 vertical to 5 horizontal; what is the whole amount of the pressure of the water against 50 feet of its length, when the depth of the water is 12 feet? *Ans.* 243,750 lbs.

2. In an empty dock the water is level with the sill at the lowermost edge of the dock gate, the level of the water on the opposite side of the gate being 10 feet above the sill. The dock gate is 10 feet wide, find the pressure in pounds on the dock gate. If the water were at a level of 5 feet in the dock, the level outside being the same as before, how much would the pressure on the gate be relieved? *Ans.* 31,250 lbs.; 7,812·5 lbs.

3. State Archimedes' principle. A cylindrical can is 6 inches in diameter and 30 inches deep, and is required, when empty, to stand in a bath of water 30 inches deep without being lifted up. To what weight must the can be loaded, the weight of a cubic foot of water being $62\frac{1}{2}$ lbs.? *Ans.* 30·679 lbs.

4. A rectangular tank, 4 feet square, is filled with water to a height of 3 feet. A rectangular block of wood, weighing 125 lbs., and having a sectional area of 4 square feet, is placed in the tank, and floats with its sides vertical and with this section horizontal. How much does the water rise in the tank, and what is now the pressure on one vertical side of the tank? *Ans.* $1\frac{1}{2}$ inches; 1,220·6 lbs.

5. The total force in the direction of the motion of a piston is the cross-sectional area of the cylinder multiplied by the pressure. Why is this so, the piston not having a plane surface?

6. An escape valve, loaded partly by a weight and partly by a spring, is fitted to a main conveying water under pressure, and is required to open automatically when the water pressure rises above a certain amount. Sketch and describe the construction of such a valve when arranged on the double beat principle, and explain clearly the hydrostatic principle involved therein.

7. Prove that when a thin spherical shell is exposed to the bursting pressure of gas or liquid the stress in the material is half as great as that within the curved surface of a thin cylindrical shell exposed to the like pressure, each shell being of the same thickness and diameter.

8. Sketch a combined plunger and bucket pump with index of parts, explaining its use and action, also sketch its application to the lifting of water from deep mines. Suppose a pump raises 5,000 gallons every half-minute from a depth of 600 feet with 30 per cent. loss in system, what is the horse-power required? *Ans.* 2,597 H.P.

9. Describe a force pump for supplying water to the accumulator of hydraulic cranes. Sketch a section through the plunger and valves.

10. Sketch and describe a force pump having a solid plunger, showing the construction of the valves. The diameter of the plunger is $2\frac{1}{2}$ inches, and it is driven by a crank 2 inches in length making 30 revolutions per minute. Find the cubic inches of water pumped in 5 minutes. *Ans.* 2,945 cubic inches.

11. Describe and illustrate by a longitudinal section, and such other views as may be necessary, the construction and action of a double-acting pump

and its valves, supposing the pump cylinder to be of $3\frac{1}{2}$ inches internal diameter, and to work at a pressure of 700 lbs. per square inch. Of what materials would the several parts be constructed?

12. A vertical single-acting pump has to elevate water 50 fathoms. The bore of the pump is 6 inches; stroke, 6 feet; number of up strokes, 10 per minute. Find (*a*) the pressure per square inch on the pump bucket when it is at the bottom of its stroke; (*b*) the weight of water discharged per minute; (*c*) the horse-power of the engine required to drive the pump, supposing 30 per cent. of the engine-power to be lost by friction, &c. Sketch an arrangement of the kind. *Ans.* (*a*) 130·28 lbs. per square inch; (*b*) 736·31 lbs.; (*c*) 9·56 H.P.

13. A 4-ton hydraulic lifting jack has a lifting ram of 2 inches in diameter, and a pump plunger of 1 inch in diameter. The jack is worked by a lever handle, the leverage being 16 to 1. What pressure must be applied at the end of the handle in order to lift a load of 25 cwts., if the efficiency of the machine is 80 per cent.? Make a vertical section of the jack, showing the valves and the mode of connecting the lever with the pump plunger. How can the weight be lowered slowly and regularly without jerks? *Ans.* 54·37 lbs.

14. Explain, with the aid of a sectional sketch, the action and construction of the hydraulic jack. How is the pressure taken off and the load slowly lowered? If the ram is 2 inches in diameter, the pump plunger $\frac{7}{8}$ inch, and the mechanical advantage of the handle 10, what is the total mechanical advantage, neglecting friciion? *Ans.* 52·26 : 1.

15. Make a sketch of a 10-ton hydraulic jib crane in which the lifting cylinder is carried in the pillar or post of the crane. What would be the diameter of ram required in the arrangement you adopt, supposing water to be supplied to the crane at a pressure of 700 lbs. per square inch, neglecting friction, &c.? *Ans.* Diameter = 6·37 inches.

16. Sketch in section the cylinder, ram, and leather collar of an hydraulic press. Explain the principle of the press and the manner in which the escape of water is prevented. Example—The sectional area of the plunger of the force pump is $\frac{1}{50}$ that of the ram, and the leverage gained by the pump handle is 12 to 1, find the pressure on the ram when a force of 60 lbs. is exerted at the end of the pump handle. *Ans.* 36,000 lbs.

17. Describe, with a sectional sketch, a hydraulic press where the ram is actuated in both directions. Show the position and forms of the cup leathers.

18. The return stroke in an hydraulic press is often accomplished by forming the ram like a piston with a very large piston-rod. Sketch in longitudinal section such a press, showing the arrangement of the leathers. What will be the relative speeds of the forward and return strokes of the ram when the larger and smaller diameters are 15 inches and 14 inches respectively, the pumps for the supply of water running at the same speed in both cases?

19. In some hydraulic presses a single valve, held down by a lever and weight, is used both to indicate and relieve the pressure. Sketch the valve in position and explain its action.

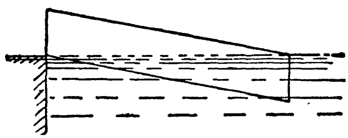
20. Describe clearly and show with sketches the construction and action of any one form of portable hydraulic riveting machine with which you are acquainted. Show clearly the valves and connections by which the pressure is applied to close the rivet, and how the pressure is released and the tool withdrawn from the rivet head when the riveting is completed. How is the water pressure conveyed to the machine?

LECTURE I.—I.C.E. QUESTIONS.

1. A rectangular sluice, 4 feet square, turns on a horizontal hinge at its top edge, the hinges being immersed 2 feet. Find the total force on the sluice and also the horizontal force which must be applied at the bottom edge of the door in order to keep it closed against this pressure. *Ans.* 3,994 lbs. of sluices; 2,330 lbs. horizontal.

2. Describe, with sketches, a hydraulic jack. The diameter of the ram of a hydraulic jack is $1\frac{3}{4}$ inch, the diameter of the plunger is $\frac{7}{8}$ inch, and the mechanical advantage of the handle is 10. Find the total mechanical advantage. *Ans.* 40. (I.C.E., *Oct.*, 1903.)

3. A uniform raft of rectangular section, which, when floating freely, is immersed to two-thirds its depth, has one end stranded so that the lower edge is in the place of flotation, as in figure. If the ends be assumed



vertical in the position of equilibrium, show that the upper edge of the sea end is also in the plane of flotation, and that the pressure on the wall end is W/4, where W is the weight of the raft.

4. A lock gate is 35 feet wide, and the heights of the water above the bottom of the gate on the two sides are 26 and 13 feet respectively. Find the resultant pressure and the height, measured from the bottom of the gate, at which it acts, the weight of water per cubic foot being 64 lbs. *Ans.* 253·6 tons at 10·1 feet from bottom.

5. Describe the action of a bucket-and-plunger pump. What is meant by "slip" of a pump, and how can it be reduced? What are the other causes of loss of energy in such a pump?

6. Give a sketch and explain the method of action of the pulsometer pump.

7. A hydraulic crane is required to lift 4 tons through a height of 40 feet; the maximum travel of the ram is 8 feet, and the water pressure 700 lbs. per square inch. Determine a suitable diameter for the ram and sketch, and describe the arrangement of the cylinder and multiplying gear. *Ans.* 9 inches diameter.

8. Describe, with sketches, a hydraulic accumulator. The ram of an accumulator has a stroke of 20 feet, and the pressure in the cylinder is 1,000 lbs. per square inch. The capacity of the accumulator is 3 H.P.-hours Find the diameter of the ram and the load on the carriage. *Ans.* 19·5 ins. diameter; load = 132·6 tons. (I.C.E., *Oct.*, 1904.)

9. A diving bell, of uniform horizontal section, 10 feet high, and which initially is full of air at atmospheric pressure, is sunk in water so that its top is 100 feet below the surface. If the barometric pressure be equivalent to 34 feet of water, find the height to which the air will ultimately rise inside the bell. *Ans.* 7·5 feet.

10. Define what is meant by the centre of pressure on a flat immersed surface. A tank, the plan of which is a square of 10 feet side, has a plane rectangular bottom inclined at 30° to the horizontal. If the minimum depth of water in the tank be 10 feet, estimate the total pressure on the

bottom of the tank and the depth of the centre of pressure below the surface of the water.

11. A dock entrance is closed by a caisson, the depth of water being 24 feet over the level floor. The width of entrance is 40 feet at the floor and 48 feet at the water-level, the side walls having a straight batter. Find the total horizontal force due to hydrostatic pressure upon the caisson. Find the true centre of action of the horizontal pressure. *Ans.* 768,000 lbs.; 15·75 feet from free surface.

12. Two pontoons are designed with the same breadth of 12 feet, and the same draught of 6 feet: the one having an immersed section of rectangular form 12 feet × 6 feet, while the immersed section of the other is a semicircle. Find the position of the metacentre in each case.

13. Sketch carefully and describe some form of valve for controlling the supply and exhaust to a hydraulic crane or press. The packing arrangements are to be clearly shown.

14. Make careful sketches showing in detail the construction of some form of double-acting, reciprocating pump, suitable for pumping water against a pressure of 700 lbs. per square inch or higher. The packing arrangements and the details of the valves are to be clearly shown. (I.C.E., *Feb.*, 1905.)

15. A ship sinks 2 inches on entering a river and then rises $1\frac{1}{2}$ inches on discharging 40 tons of cargo. Find its original displacement. Specific gravity of sea water = 1·025.

16. Describe how water under pressure is caused to do work; and sketch a satisfactory form of hydraulic lift.

17. State the law governing the transmission of pressure by fluids. In a hydraulic jack, the diameters of the ram and the plunger are 3 inches and 1 inch respectively, the lever arm of the force about the fulcrum is 20 times that of the plunger, and the efficiency is 70 per cent. Calculate the load lifted by a force of 50 lbs. What is the pressure in the hydraulic cylinder? (I.C.E., *Oct.*, 1906.)

18. A tidal flap, uniform in section and weighing 2,000 lbs., is 4 feet wide and 6 feet deep. It hangs on a horizontal hinge through the top edge, at an angle of 60° with the horizontal. Assuming that the water level on the outer side is 0·26 foot, measured vertically, below the hinge, find the water level on the inner side, relative to the hinge, when the flap is just about to open.

19. A diving bell is in the form of a hollow circular cylinder 8 feet high inside, open at the bottom and closed at the top. When the axis is vertical and the bottom just touches the water surface, it is filled with air at atmospheric pressure, 15 lbs. per square inch, and at a temperature of 60° F. It is lowered gradually into the water, the axis remaining vertical, until its bottom end is 20 feet below the surface. Find the height to which the water rises inside the bell, and the pressure of the air inside, the temperature remaining the same. Find also the air pressure in lbs. per square inch which must be supplied to the bell to keep the water out.

20. Find an expression for the depth below the surface of the centre of pressure on one side of a vertical plane immersed in a fluid. A circular plate is placed vertically in water, and just touches the surface. Its diameter is 3 feet. Find the depth of the centre of pressure on one side, having given that the second moment or moment of inertia of a circle about a diameter is $\frac{\pi}{64}d^4$. (I.C.E., *Oct.*, 1907.)

LECTURE II.

CONTENTS.—Frictional Resistances and Efficiencies of Machines in General —Example I.—Application to the Steam Engine—Efficiency of a Reversible Machine—Example II.—Movable Hydraulic Cranes—Movable Electric Crane—Details of Hoisting Brake and Levers for Working the 3-Ton Electric Crane—Abstract of Report, Tables and Curves on Comparative Trials for Efficiency of 3-Ton Hydraulic and Electric Cranes—Relative Cost of Hydraulic and Electric Power—Crane Tests—Explanation of Efficiency Curves—Electric Cranes for Manchester Ship Canal—Questions.

Frictional Resistances and Efficiencies of Machines in General.— In Lecture VII., Vol. I., we showed how to calculate the work lost in friction in the cases of plane and cylindrical surfaces. As these are the principal kinds of rubbing surfaces in machinery, we might, if we knew the exact pressures between the various surfaces, calculate the total frictional losses and thus find the efficiency of the machine. But there are other losses of work during its transmission, such as the bending of ropes, belts, chains, &c., which it is almost impossible to calculate with exactness. Even if we could calculate all these various losses, the task would be a most tedious and unprofitable one. Hence, in finding the efficiency of any machine, we have recourse to direct experiment on the machine as a whole, the results of which furnish us with data from which we can determine the exact efficiency. In general, however, a machine will not have the same efficiency when working under different loads, owing to the fact that the frictional and other resistances are not proportional to the efforts and loads. To make this clearer, suppose we take the case of a steam engine. Here the friction between the piston and its cylinder, the piston-rod, valve rods, &c., and their stuffing boxes, as well as the friction at the journals due to the weights of the flywheel and shaft, are, severally, constant in amount, whether the engine be developing full power or running light. Hence, in all machines there is a certain proportion of the frictional and other resistances constant, whatever be the magnitude of the effort and the load. This (as just explained in the case of the steam engine) is due to forces between the parts of the machine itself, such as the weights and inertia of the moving parts, or the resistances offered to the bending and stretching of parts, and which have little or no connection with the effort exerted or the load overcome. From these facts we see that the lost work will be proportionally less, and the useful work proportionally more the greater the total work expended. In other words, *the efficiency of a machine will, in general, be higher the greater the load.* This statement, however, is not true for hydraulic and other machines where fluid resistances occur, or where the speed of the machinery is *very* great. As will

be seen later on, fluid resistances increase very rapidly with the speed. In high-speed machinery the effects of the inertia of the moving parts introduce other serious losses which must not be ignored when calculating their efficiency.

The "Principle of Work," when applied to a machine, has already been written in the form :—

Total work expended = Useful work done + Lost work.

Or, $$W_T = W_U + W_L \quad \text{[see eqn. (I.), Lect. IV.]}$$

The last term, W_L, is made up of two distinct parts—one part depending on W_T and W_U, and a second part which is constant, and, therefore, independent of W_T and W_U. Hence, we may put :—

$$W_L = \mu_1 W_T + \mu_2 W_U + C.$$

Where μ_1 μ_2 are numerical coefficients, and $\mu_1 W_T$, $\mu_2 W_U$ are the frictional resistance due to the effort and load applied; while C represents an amount of lost work which is constant for the same machine

$$\therefore \quad W_T = W_U + \mu_1 W_T + \mu_2 W_U + C.$$

Or, $$(1 - \mu_1) W_T = (1 + \mu_2) W_U + C. \quad \ldots \ldots \quad \text{(I)}$$

This is a *general* equation for the "Principle of Work" as applied to machines. In most machines the coefficient, μ_1, which depends on the effort, Q, must necessarily be very small, and may sometimes be neglected.

Dividing both sides of the equation (I) by $(1 - \mu_1)$ we get :—

$$W_T = \frac{1 + \mu_2}{1 - \mu_1} W_U + \frac{C}{1 - \mu_1},$$

$$\text{,,} \quad = \left(1 + \frac{\mu_1 + \mu_2}{1 - \mu_1}\right) W_U + F.$$

Or, $$W_T = (1 + f) W_U + F. \quad \ldots \ldots \quad \text{(II)}$$

Where $f = \left(\frac{\mu_1 + \mu_2}{1 - \mu_1}\right)$ and $F = \left(\frac{C}{1 - \mu_1}\right)$ are new constants derived from the old ones, as shown.

For some purposes, it is more convenient to write equation (II) in the following forms :—

$$W_T = k W_U + F. \quad \ldots \ldots \quad \text{(III)}$$

Or, $$Q x = k W y + F. \quad \ldots \ldots \quad \text{(IV)}$$

Where (as in Lecture IV., Vol. I.) x and y are respectively the displacements of the effort Q and the load W, in a given time.

Suppose the machine to run light. Then $W = 0$, and Q_0 is the effort required to drive the machine.

From equation (IV), we get :—

$$Q_0 x = F.$$

That is, F *represents the work done in driving the machine unloaded.*

Dividing both sides of this equation by x, we get :—

$$Q_0 = \frac{F}{x}.$$

This is the effort required to drive the machine light.

Substituting the above value for F, in equation (IV), we get :—

$$Q x = k \; W y + Q_0 x.$$

Or,
$$Q = k \; W \frac{y}{x} + Q_0$$

$\therefore$
$$Q = k \; W \frac{v}{V} + Q_0. \qquad \text{(V)}$$

This is a *general* equation connecting the effort Q and the load W for any machine. Since the velocity ratio, $\frac{V}{v}$, is constant for the same machine, we might write the last equation in this convenient form :—

$$Q = K \; W + Q_0. \qquad \text{(VI)}$$

Where, $K = k \Big/ \frac{V}{v}$, and can be found by experiment for any machine.

Example I.—In an ordinary block and tackle having three sheaves in the upper and two in the lower block, it is found by experiment that a force of 11 lbs. is required to lift a weight of 40 lbs., and a force of $24\frac{1}{2}$ lbs. to lift a weight of 100 lbs. Find a general expression for the relation between Q and W in this arrangement, and the weight which could be raised by a force of 56 lbs. Find, also, the efficiency of the machine in all three cases, and the actual mechanical advantage.

Answer.—The general relation between Q and W must be of the form :—

$$Q = K \; W + Q_0. \qquad (1)$$

From the results of the first experiment, we get :—

$$11 = 40\,K + Q_0 \quad \ldots \ldots \ldots \quad (2)$$

And from the results of the second experiment, we get :—

$$24\tfrac{1}{2} = 100\,K + Q_0. \quad \ldots \ldots \ldots \quad (3)$$

(3)–(2) $\quad 13\tfrac{1}{2} = 60\,K,$

$$\therefore \quad K = \frac{27}{120} = \frac{9}{40}.$$

Substituting this value of K in equation (2), we get :—

$$11 = \frac{9}{40} \times 40 + Q_0,$$

$$\therefore \quad Q_0 = 2 \text{ lbs.}$$

Or, the effort required to drive the machine light is 2 lbs.

Substituting the values of K and Q_0 in equation (1), we get :—

$$Q = \frac{9}{40}\,W + 2 \quad \ldots \ldots \ldots \quad (4)$$

which is the general formula required.

To find the weight which could be lifted by an effort of 56 lbs., we substitute this value in equation (4), and get :—

$$56 = \frac{9}{40}\,W + 2,$$

$$\therefore \quad W = \frac{54 \times 40}{9} = 240 \text{ lbs.}$$

The efficiency of the machine in any case is found from the usual formula, viz. :—

$$\textit{Efficiency} = \frac{\text{Useful work done}}{\text{Total work expended}} = \frac{W\,y}{Q\,x} = \frac{W}{Q}\,\frac{v}{V}.$$

Since there are five ropes supporting the weight, W, in this system of block and tackle, it is clear that :—

$$\frac{v}{V} = \frac{1}{5}.$$

$$\therefore \quad \text{Efficiency} = \frac{1}{5}\,\frac{W}{Q}.$$

When raising the weight of 40 lbs., we get :—

$$\text{Efficiency} = \frac{1}{5} \times \frac{40}{11} = \cdot727, \text{ or } 72\cdot7 \text{ per cent.} \quad (1)$$

When raising the weight of 100 lbs., we get :—

$$\text{Efficiency} = \frac{1}{5} \times \frac{100}{24\cdot5} = \cdot816, \text{ or } 81\cdot6 \text{ per cent.} \quad (2)$$

When raising the weight of 240 lbs., we get :—

$$\text{Efficiency} = \frac{1}{5} \times \frac{240}{56} = \cdot857, \text{ or } 85\cdot7 \text{ per cent.} \quad (3)$$

The student should notice how the efficiency increases as the load, W, is increased.

$$\left.\begin{array}{l}\textit{The actual advantage}\\ \textit{when raising } 40 \textit{ lbs.}\end{array}\right\} = \frac{W}{Q} = \frac{40}{11} = 3\cdot63.$$

By multiplying the numbers expressing the efficiencies by 5 (the number of ropes attached to the lower block), we get the actual mechanical advantage in each case.

Application to the Steam Engine.—Since the above reasoning is applicable to *all* machines, when the frictional resistances are not greatly influenced by speed, &c., we may here show its application to the steam engine.

Let p_m = Mean pressure on piston in lbs. per square inch.
" p_u = Mean pressure per square inch (being part of p_m) required to overcome the **useful load**, W.
" p_f = Mean pressure per square inch required to drive the engine when unloaded, or simply the "friction pressure" per square inch.

Since, $\frac{V}{v}$ or the velocity ratio does not enter into this case, and all the pressures are considered as acting on the same piston, we get from equation (V) :—

$$p_m = k\,p_u + p_f.$$

By experiment it is found that the constant p_f, called the "Friction Pressure," has a value between 1 and $1\frac{1}{2}$ lbs. per square inch, in ordinary land engines, and about 2 lbs. per square inch in marine engines. The value of k is about 1·15, but varies with both size and speed of engine. In large or high-speed engines, k is often less than 1·15, though it can never be less than 1.

Efficiency of a Reversible Machine.—The student will have noticed, from the Table of Efficiencies in Lecture II., Vol. I., the great difference in the efficiencies of such machines as the ordinary block and tackle and the Weston's pulley block. In the examples worked out at the end of Lecture IV., Vol. I., it was proved that the efficiency of the former machine may be as high as 75 per cent., while the efficiency of the latter never reaches 50 per cent., and seldom exceeds 40 per cent. He also knows that when the efficiency of any machine is less than 50 per cent. it will not reverse, even if the hauling force or effort be withdrawn. Hence the difference in the working of the two machines just mentioned. The "block and tackle" is, under ordinary conditions, a *reversible* machine (*i.e.*, the load at the lower block is capable of overcoming a smaller load at the hauling part of the rope), while the Weston's pulley block will not reverse even when the hauling force is entirely withdrawn. To lower the load with a Weston's block a force has to be applied to the opposite part of the hauling chain from that at which the effort had to be applied when raising the load.

The screw, wedge, and worm-wheel arrangements are, generally speaking, examples of non-reversible machines. In fact, their usefulness depends to a large extent on this condition.

We can now show that the efficiency of a machine, when working reversed, is not the same as when working in the usual way. Further, if the efficiency of any machine be less than ·5 or 50 per cent., it is not reversible.

We have seen that in any direct working machine the "Principle of Work" takes the form :—

$$(1 - \mu_1)\, W_T = (1 + \mu_2)\, W_U + C. \text{ [equation (I)]}$$

Or, $$(1 - \mu_1)\, Q\, x = (1 + \mu_2)\, W\, y + C \quad . \quad . \quad . \quad . \quad (1)$$

where the coefficients, μ_1 and μ_2, have the meanings already assigned to them, and C represents a quantity of work absorbed in the machine, but which is independent of both Q and W.

Now, suppose we gradually diminish the effort, Q, until the machine reverses. When this takes place, let the new value of Q be denoted by w, so that the new load is w, and the original load, W, becomes the new effort. The above relation being still approximately true, we only require to substitute the new values for the new effort and load. At the same time it must be observed that the coefficients, μ_1 and μ_2, are taken along with their proper terms, $w\, x$ and $W\, y$; *i.e.*, $\mu_1\, w\, x$ is the lost work due to new load, w, while $\mu_2\, W\, y$ is lost work due to new effort or original load, W.

Then :—

$$(1 - \mu_2)\, W\, y = (1 + \mu_1)\, w\, x + C. \quad . \quad . \quad . \quad . \quad (2)$$

Now subtracting equation (2) from (1), in order to eliminate C, we get :—

$$(1 - \mu_1)\, Q\, x - (1 - \mu_2)\, W\, y = (1 + \mu_2)\, W\, y - (1 + \mu_1)\, w\, x$$

$$\therefore \qquad (1 + \mu_1)\, w\, x = 2\, W\, y - (1 - \mu_1)\, Q\, x.$$

Dividing both sides by $(1 + \mu_1)$ and $W\, y$, we get :—

$$\frac{w\, x}{W\, y} = \frac{2}{1 + \mu_1} - \left(\frac{1 - \mu_1}{1 + \mu_1}\right) \frac{Q\, x}{W\, y}.$$

$$\textit{Efficiency when reversed} = \frac{\text{Useful work done in raising } w}{\text{Total work expended by W}}.$$

$$\text{,,} \qquad = \frac{w\, x}{W\, y},$$

$$\text{,,} \qquad = \frac{2}{1 + \mu_1} - \left(\frac{1 - \mu_1}{1 + \mu_1}\right) \frac{Q\, x}{W\, y}.$$

But $\dfrac{W\, y}{Q\, x}$ = the *original efficiency* of the machine, or efficiency when working in the usual manner.

$$\therefore \quad \left.\begin{array}{c}\text{Efficiency of Machine}\\ \text{when Reversed}\end{array}\right\} = \frac{2}{1 + \mu_1} - \left(\frac{1 - \mu_1}{1 + \mu_1}\right) \times \frac{1}{\eta} \quad . \quad \text{(VII)}$$

where η denotes the original efficiency of the machine.

It is clear that the machine will not reverse unless the above efficiency be greater than 0—

$$\textit{i.e.}, \text{ unless} \quad . \quad . \quad \frac{2}{1 + \mu_1} - \left(\frac{1 - \mu_1}{1 + \mu_1}\right) \times \frac{1}{\eta} > 0.$$

$$\textit{i.e.}, \text{ unless} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \frac{2}{1 + \mu_1} > \left(\frac{1 - \mu_1}{1 + \mu_1}\right) \times \frac{1}{\eta}.$$

$$\textit{i.e.}, \text{ unless} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \eta > \tfrac{1}{2}\,(1 - \mu_1).$$

Consequently, the machine will not reverse until the original efficiency, η, be greater than $\frac{1}{2}\,(1 - \mu_1)$, and, if it be less than this, it will not reverse even if the original hauling force, Q, be entirely withdrawn. For, if $\eta = \frac{1}{2}\,(1 - \mu_1)$, the efficiency of the machine when reversed would vanish (as may be seen by substituting this value in equation (VII)). If $\eta < \frac{1}{2}\,(1 - \mu_1)$, the efficiency would be negative, which is absurd.

We have already stated that in most machines the fraction, μ_1, is *very small*, and may be neglected. Hence we get the important statement that

A machine will not reverse even when the hauling force is entirely withdrawn, if the efficiency is less than ½ or 50 per cent.

EXAMPLE II.—Apply the "Principle of Work" to calculate the relation between P and W in the screw lifting jack, fitted with screw and worm-wheel gear. The handle which works the jack has a radius of 14 inches, pitch of screw 1 inch, number of teeth on worm-wheel 20, and the worm is double threaded; find the force which must be applied at the end of the handle in order to raise a weight of 4 tons, friction being neglected. If the actual force required to raise this weight be 40 lbs., what is the efficiency of the apparatus ?

ANSWER. — Let L = length of handle, p = pitch of screw, N = number of teeth on worm-wheel, n = number of threads on worm.

(1) Suppose the handle to make one complete turn. Then, since there are n threads on the worm, and N teeth on the worm-wheel, it is clear, that for one turn of the handle, the worm-wheel, which forms the nut of the screw, will have made $\frac{n}{N}$ part of a complete turn. Hence the weight, W, will have been raised through a height $\frac{n}{N} \times p$.

∴ By the Principle of Work, we get :—

P × its displacement = W × its displacement.

$$\therefore \qquad P \times 2\pi L = W \times \frac{n}{N} \times p,$$

$$\therefore \qquad \frac{P}{W} = \frac{np}{2\pi L N}.$$

(2) In the example $p = 1''$, $L = 14''$, $n = 2$, $N = 20$, $W = 4 \times 2{,}240$ lbs.

$$\therefore \qquad P = 4 \times 2{,}240 \times \frac{2 \times 1}{2 \times \frac{22}{7} \times 14 \times 20} \text{ lbs.,}$$

$$,, = 10{\cdot}18 \text{ lbs.}$$

(3) $\qquad$ ***Efficiency*** $= \frac{P}{Q} = \frac{10{\cdot}18}{40} = {\cdot}2545$, **or 25·45 per cent.**

Movable Hydraulic Cranes.—The following extracts from the "General Specification" by Mr. George H. Baxter, Chief Mechanical Engineer to the Clyde Navigation Trustees, are herewith reproduced along with the accompanying reduced drawings, in order to enable the student to form an idea of the main requirements fulfilled by these double-power 5 and 3-ton hydraulic cranes at the Prince's Dock, Glasgow. The cranes were made by A. Chaplin & Co., Mechanical Engineers, Govan, and have been constantly in use since 1898. Interesting comparative tests were made in March, 1903, between one of these cranes working with its 3-ton lifting ram, and a 3-ton electrical crane, which are described immediately after their specifications. These tests afford useful information regarding the two systems of power supply :—

General Description of Cranes.—"Each crane to be mounted on a steel travelling carriage, with arch for passage of locomotives, having a rail gauge of 14 feet between centres of double rails, which are to be provided and laid on quay and cope of quay wall by the Trustees. Jib to be capable of revolving through at least one and a-quarter turns. The cranes to be double-powered, to lift maximum loads of 5 tons and 3 tons. All the lifts to be by single chain, and the working pressure to be 750 lbs. per square inch. The cranes to have a factor of safety throughout of not less than 8 to 1, excepting the hoisting chains."

Carriages.—"The carriages to be supported by four wheels of cast iron, bushed with gun-metal ⅝-inch thick, with wrought-steel weldless tyres shrunk on each side of central flange after being turned and bored. The axles to be turned steel forgings, secured by strong split cotters. Brackets carrying wheels to be so arranged that the wheels can be easily removed. Shackles and steadying screws to be provided at each corner. The outer side of carriage to be kept at least one foot back from edge of cope, and a hand-rail to be run (the whole length of carriage) outside for convenience in shifting mooring ropes."

Range of Lift.—"Each of the cranes to have a total range of lift of 75 feet, being 35 feet above the cope to 40 feet below, and the chain to project 32 feet beyond face of quay wall when the jib is square to it."

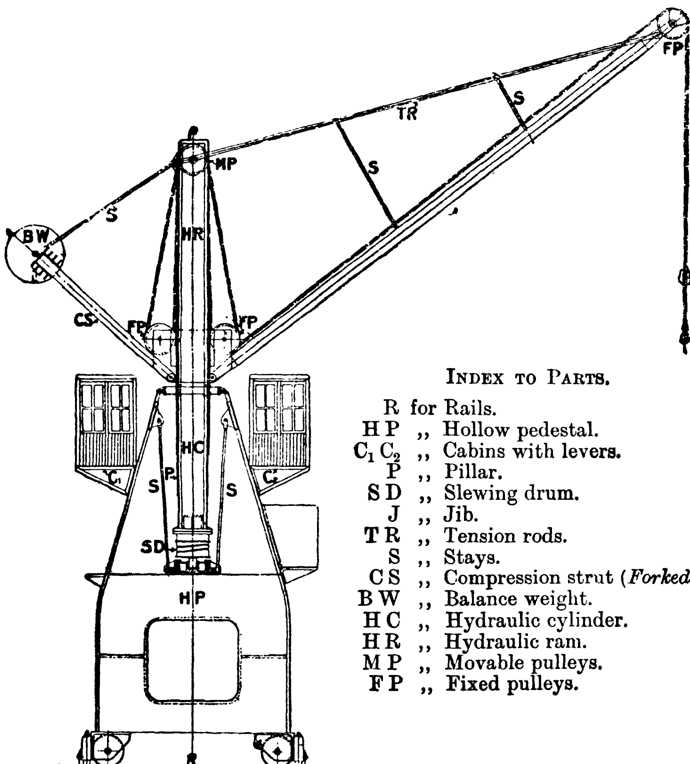
Jib and Counter-balance Weight.—"The jib of each crane to have a rake of not less than 41 feet, and the jib-head pulley to be 60 feet from level of cope of quay wall. Jib stays to be secured independently of pins on which jib-head pulley revolves, so that the latter may be taken down without disturbing the stays. A steel bracket on hinged joint, or other means, to be provided for locking jib when cranes are not in use. Wrought-iron ladder to be fitted to each jib to give access for lubricating jib-head pulley. Counter-balance weight to be high enough to clear roof of shed."

Houses for Driver.—"A house to be provided for the driver on each side of upper part of carriage. Handles for working to be conveniently placed, and marked for hoisting, lowering, and slewing."

Platforms, Hand-rails, Ladders, Guards, &c.—"Platforms, hand-rails, gangways, ladders, footsteps, &c., to be fitted. Suitable lock-up doors and other arrangements to be made to give access to all chains, pulleys, bearings, and other parts, for lubrication and repair. Guards for chains and rollers or rubbing pieces to be fitted wherever required. Elm sheathing, 1½ inches thick, to be laid on inside of jib, and secured thereto by bolts and nuts."

Chains.—"The hoisting chains to be 1 inch diameter and tested in accordance with the standard requirements of the Clyde Navigation Trustees. Each hoisting chain to have a swivel hook of special form and strength, shown by the drawing. A balance ball to weigh at least $2\frac{1}{2}$ cwts., bolted on in halves a few feet above the hook. Also, a wrought-iron or steel clip of ample strength, bolted to standing part of chain, for adjusting height of hook."

Pulleys.—"The standing pulleys to be of the diameter shown in the



INDEX TO PARTS.

R for Rails.
HP ,, Hollow pedestal.
C_1 C_2 ,, Cabins with levers.
P ,, Pillar.
SD ,, Slewing drum.
J ,, Jib.
TR ,, Tension rods.
S ,, Stays.
CS ,, Compression strut (*Forked*).
BW ,, Balance weight.
HC ,, Hydraulic cylinder.
HR ,, Hydraulic ram.
MP ,, Movable pulleys.
FP ,, Fixed pulleys.

GENERAL VIEW OF MOVABLE HYDRAULIC CRANE.
(Made by A. Chaplin & Co., Govan, for the Clyde Trust.)

drawings, and to be arranged on either side of ram as in hydraulic cranes at Queen's Dock. Suitable guards for chain to be provided. Pins for pulleys and carriage wheels to be bored longitudinally, with oil cup in one end and oil hole to each pulley and wheel. All pulleys to be bushed with gun-metal, and be fitted with two steady pins tapped through boss."

Valves.—"The working valves for hoisting, lowering, and slewing to be of the conical description, with levers and weights."

Pipes and Cocks.—"All pipes exposed to the full working pressure to be of strong lap-welded hydraulic tubing of the required thickness, with flanges of cast-iron, turned and faced, with recessed joints to standard sizes. Drain cocks of gun-metal to be fitted to all hoisting and slewing cylinders and all pipes at lowest parts, so as to thoroughly drain them. Air cocks to be fitted to all cylinders and pipes as high as practicable."

Travelling Gear.—"Efficient hand travelling gear to be provided on each crane, so that it may be quickly and easily transported along the quay by not more than six men."

Gas Fittings.—"To prevent damage by frost, each crane to be provided with two gas stoves and burners, including wrought-iron pipes, cocks, and other fittings, for hoisting and slewing cylinders, pipes, &c. A suitable gas stove, with all necessary connections, to be supplied in each crane house. Wrought-iron gas piping, with cock and connection for india-rubber tubing, to be fitted on each carriage, for supplying stoves, &c."

Movable Electric Crane.—The following extracts from the "General Specification," by George H. Baxter, Chief Mechanical Engineer to the Clyde Navigation Trustees, are herewith reproduced, along with the accompanying reduced drawings, in order to enable the student to form an idea of the main requirements fulfilled by the 3-ton movable electric crane, as built and erected at Prince's Dock, Glasgow, by Stothert & Pitt, Bath.* After the specification there follows the data of the comparative tests between this crane and the previously described hydraulic one:—

General Description.—"The crane is to be movable, mounted on a high portal carriage to run on a line of double rails, 14 feet apart between centres, laid along the front of the South Quay, Prince's Dock, Glasgow Harbour. The crane is intended to be used for carrying out an extensive series of tests for the purpose of comparing the relative efficiencies of electric and hydraulic cranes of similar capacity under varying conditions and over a prolonged period. Provision is, therefore, to be made for applying and fitting up suitable recording instruments for the measurement of electric energy delivered to the hoisting and slewing motors, and of the mechanical energy developed by the crane."

General Dimensions.—"The following are the general dimensions of the crane:—

Maximum working load, ...	3 tons.
Total range of lift, ...	80 feet.
Radius of lifting rope, ...	41 ,,
Projection of rope beyond face of quay, . . .	32 ,,
Centre of jib-head pulley above cope, . . .	60 ,,
Clear lift above cope of quay, ...	50 ,,
Extreme radius of any revolving part at back of crane, .	9 ,,
Clear height from cope to lowest revolving part of crane,	29 ,,

Speeds.—"The lifting speed with a load of 3 tons to be 150 feet per minute, the speed with smaller loads to be correspondingly greater. The slewing speed measured at the hook to be 300 feet per minute with maximum load."

* The electric motor, volt and ampere metres for this crane were made by Siemens Brothers & Co., Ltd., of Woolwich and London.

Factors of Safety and Stability.—"The carriage, crane framing and jib, gearing and wire rope, to have a factor of safety of 10 to 1. Stability to be maintained with a load of 6 tons suspended in the crane hook."

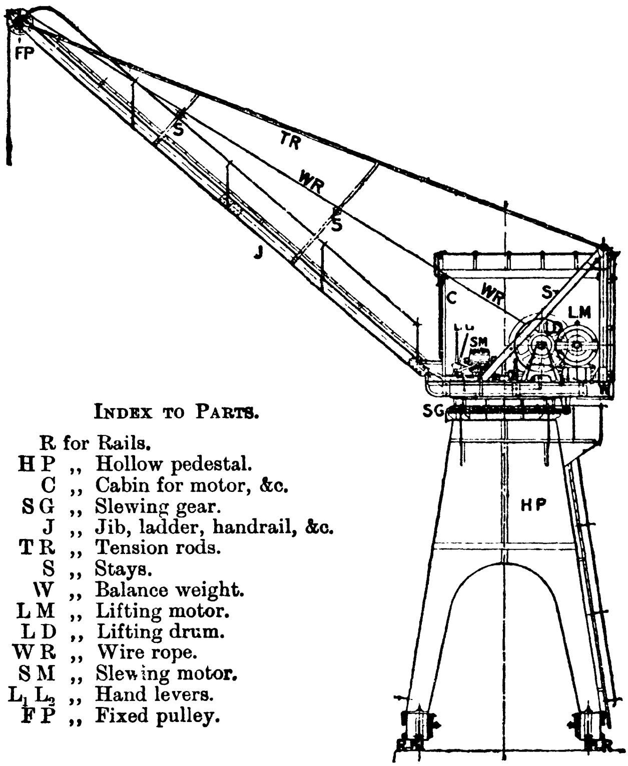
Carriage.—"The carriage is to be constructed with an arch 15 feet high, sufficiently wide for the passage of locomotives. To be made of Siemens steel plates, angles, and bars. The wheel base to be 14 feet. The carriage to be supported by four strong cast-steel wheels turned to 2 feet 6 inches diameter on tread, and having a central flange, 1 inch deep by $1\frac{3}{4}$ inch wide. The wheel bosses to be bored, faced, and bushed with gun-metal, $\frac{5}{8}$ inch thick. The axles to be turned steel forgings, secured by strong split cotters; oil holes to be drilled through centre with short bent lubricating pipes screwed into inner ends. The length and diameter of axle bearings to be specified or shown on plan. Brackets carrying wheels to be of approved design, and so arranged that the wheels can be easily removed if required. Shackles and steadying screws similar to those on hydraulic cranes to be provided at each corner. The side of carriage next dock to be kept at least 1 foot back from the edge of the quay, and a hand-rail to be run the whole length of carriage outside for convenience in shifting mooring ropes. Provision to be made on side of carriage for connecting the motors with the main circuit conductors."

Framing Jib, &c.—"The framing of the revolving part of the crane, jib, and back stays to be made of steel plates and angles, channels, or tee bars carefully fitted and riveted together. The jib stays to be steel forgings or bars drawn or shaped out of the solid without welds, and to be secured by turned pins independently of the pin on which the jib-head pulley revolves, so that the latter may be removed without disturbing the stays. An 18-inch wrought-iron ladder, with hand-rail on each side, to be fitted to jib to give access for lubricating the jib-head pulley. The pulley to be of cast steel, turned in groove to not less than 24 times the diameter of hoisting rope. To be provided with rope, guides, and guards. The pin to run in adjustable gun-metal bushes, and the bearings to have fixed lubricators of approved pattern. Counterbalance weights to be provided in back end of framing to balance the jib and the load."

Centre Post, Roller Path, and Rollers.—"A strong centre post of forged ingot steel, with hole drilled throughout its length for the conductors to pass through, to be turned all over and fitted to strong cast-iron bed plate, or sockets secured to upper part of carriage by turned and fitted bolts. If a roller path is fitted, it is to be a solid ring of Siemens steel accurately turned and faced flat, and secured to carriage by turned and fitted bolts. The rollers are to be of cast steel, truly turned, bored, and keyed to fit strong steel axles, having large bearing surfaces working in gun-metal bushes, adjustable by bolts and nuts, and to be easily removable for repairs."

Plating and Riveting.—"All plate edges are to be planed, all butts to be accurately fitted, and ends of bars neatly trimmed. Rivet holes in framing, jib, &c., are to be drilled in place, and all holes to be made fair without drifting. All rivets to be snap-headed outside."

Hoisting Winch.—"To be single-geared, driven by a separate motor. The bed-plate to be of cast iron, planed to receive the side cheeks and hoisting motor. One of the cheeks to be of box form, planed, bolted together in halves, jointed, and made oil tight; the lower half to form an oil-bath for main spur wheel and the pinion on armature shaft. The barrel to be of cast iron, 24 times the diameter of the hoisting rope, and to be turned with a continuous spiral groove, long enough to take the whole rope required without riding. The maximum load is to be taken on a



INDEX TO PARTS.

R for Rails.
H P ,, Hollow pedestal.
C ,, Cabin for motor, &c.
S G ,, Slewing gear.
J ,, Jib, ladder, handrail, &c.
T R ,, Tension rods.
S ,, Stays.
W ,, Balance weight.
L M ,, Lifting motor.
L D ,, Lifting drum.
W R ,, Wire rope.
S M ,, Slewing motor.
L_1 L_2 ,, Hand levers.
F P ,, Fixed pulley.

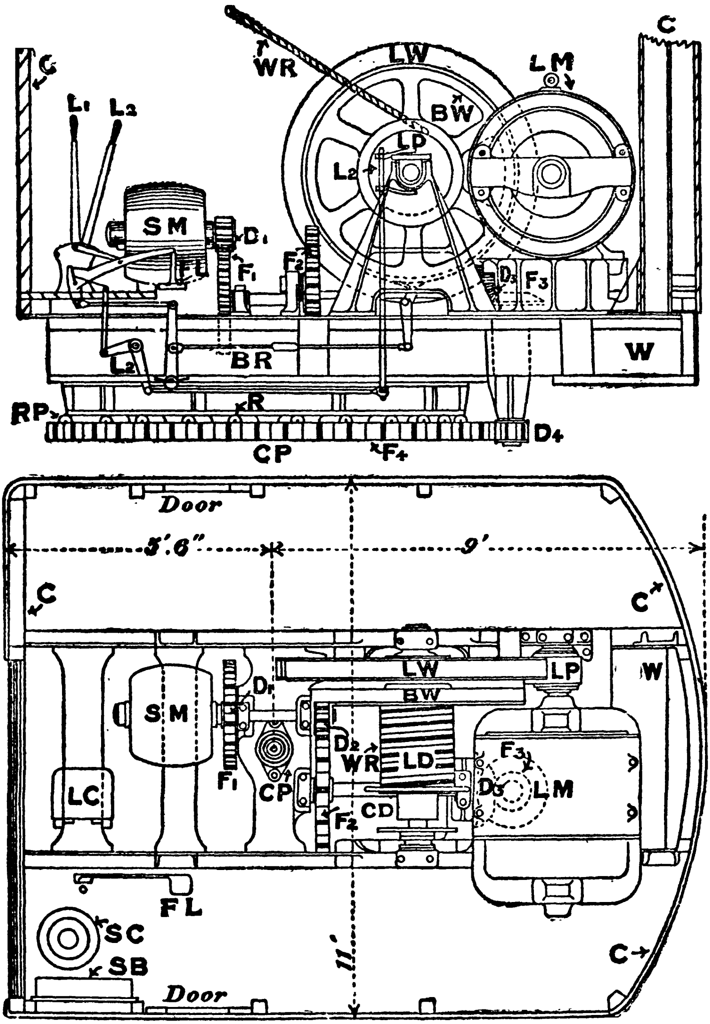
GENERAL VIEW OF A 3-TON MOVABLE ELECTRIC CRANE.
(Made by Stothert & Pitt, Bath, for the Clyde Trust.)

INDEX TO PARTS OF LIFTING, LOWERING, AND SLEWING GEAR.

(See Section and Plan on opposite page.)

C for Cabin.
S B ,, Switch board.
L_2 ,, Lever for clutch, C D.
C D ,, Coil-clutch and disc.
L C ,, Lifting controller.
L M ,, ,, motor.
L P ,, ,, ,, pinion.
L W ,, ,, ,, wheel.
L D ,, ,, ,, drum.
W R ,, Wire rope.
F L ,, Foot-brake lever.
B R ,, Brake rod.

B W for Brake wheel.
L_1 ,, Lever for slewing brake.
S C ,, Slewing controller.
S M ,, Slewing motor.
D_1 D_2 D_3 D_4 ,, Driving pinions.
F_1 F_2 F_3 F_4 ,, Follower wheels.
R P ,, Roller path.
R ,, Rollers.
C P ,, Central pivot.
W ,, Balance weight.



VERTICAL SECTION AND PLAN OF CABIN AND DRIVING GEAR
For the 3-Ton Electric Crane belonging to the Clyde Trust.

single rope. The main spur wheel is to be of cast steel, the pinion of forged steel, both to have wide teeth of suitable pitch, accurate form, and machine cut. The ratio of gearing and diameter of barrel and jib-head pulley to be shown on contractor's plan."

Hoisting Brake.—"**An automatic brake to be provided.** This brake to be worked also by a hand or foot lever, and to effectually hold the maximum load at any point. It must be so designed, that when the current is switched on to the hoisting motor, the brake is automatically released the moment hoisting begins. And, in switching off, or in the event of failure of current, the brake would be put in gear automatically, and without shock." (*See the pages describing construction and action of this brake at the end of this Specification.*)

Hoisting Rope.—"To be made of the highest quality of plough steel wire, specially flexible and free from twisting. The rope to be provided with an overhauling weight in halves, and an eye with solid wrought-iron thimble at end, bored to fit the shackle-pin; also a 3-feet length of $\frac{13}{16}$-inch short-link crane chain, and swivel hook of approved design. The ultimate tensile strength of the wire (which is to be tested and approved of before the rope is made) is not to exceed 110 tons per square inch."

Slewing Gear.—"To be worked by separate motor, having a wrought-steel worm keyed on end of armature shaft, gearing into a phosphor-bronze worm wheel on vertical shaft, with pinion at lower end in gear with the slewing wheel, which is to be secured to top of carriage with turned and fitted bolts. The worm to have large thrust bearings of gun-metal, and to run in an oil-bath with dustproof cover. The vertical shaft to be of large diameter to ensure stiffness, and to have collars forged on at ends. The bearings are to be of gun-metal, adjustable by means of bolts and nuts. All teeth to be of suitable pitch, accurate form, and machine cut. A brake worked by foot lever to be provided to control the speed at any point."

Travelling Gear.—"The crane to be provided with efficient travelling gear fitted to each carriage wheel, worked by hand, and so connected by means of clutches that it may be disengaged to admit of the crane being moved along the quay by a hydraulic capstan. Spur wheels on axles of carriage wheels to be 6 inches clear of the ground."

Electric Motors.—"Both motors are to be of the drum armature enclosed type. They are to be series-wound, and constructed to work on a continuous current circuit at an E.M.F. of 230 volts, but capable of working at 260 volts if required. The hoisting motor to be 50-brake H.P., and slewing motor 10-brake H.P. Both are to be designed to give every facility for examination and repairs. The armatures to be of the strongest form of construction, well insulated throughout, with provision for thorough ventilation. The rise of temperature in the armature must not exceed 60° F. above the surrounding atmosphere. The commutators are to be made of pure annealed copper bars, insulated with mica, and to have large wearing surfaces, with self-adjusting carbon brushes. The armature shafts to be of forged steel, working in long bearings lined with white metal, bored out perfectly smooth and true, and provided with efficient automatic lubricating apparatus of the ring type. Chambers to be formed at the ends of each bearing to hold waste oil, with gauge-cocks, drain-cocks, and pipes led to an oil reservoir."

Controlling Switches.—"These are to be of the most improved type, with graduated resistances suitable for cranes. One controlling switch is to be provided for each motor, and of such dimensions as shall prevent undue heating when the crane is working continuously for six hours. The switch contacts must be large and carefully constructed, so as to prevent sparking. These are to be enclosed in an iron casing, with covers and doors for examination and repairs. The controlling switches to be so constructed that the motors may be started, regulated, stopped, reversed, or ran at

varying speeds, with varying loads in the simplest and most reliable manner. The regulating handles are to be conveniently arranged in such a position that the driver may see the load during the whole time it is suspended in the crane, and the handles should be so connected up, that they move in the direction it is desired to move the load."

Switchboard, Switches, and Cut-outs, &c.—"A double-pole switch of the quick break lever type and two single-pole cut-outs for the hoisting and slewing motors, together with separate switches and cut-outs for the electric lighting and heating circuits, to be all arranged on a suitable switchboard of slate or marble, framed in channel iron $\frac{3}{16}$ of an inch thick lined with teak. The switches and cut-outs to be properly enclosed and protected, so as to prevent risk of fire or shock in any way, and to be conveniently placed within easy reach of the driver. Sweating sockets to be provided, to which the cables and wiring are to be connected. The cut-out terminals shall be so arranged that the fuses of flat tinfoil may be easily and securely fitted by the craneman. The securing nuts are to be of bright mild steel, and the fuse break not less than 5 inches. Each switch and instrument shall be capable of conveying the maximum current continuously, without sensible rise of temperature. The covered flexible connections and cables shall be of such area, that the current density shall not exceed 800, the gun-metal parts 400, and the effective contact surfaces of switches 200 amperes per square inch."

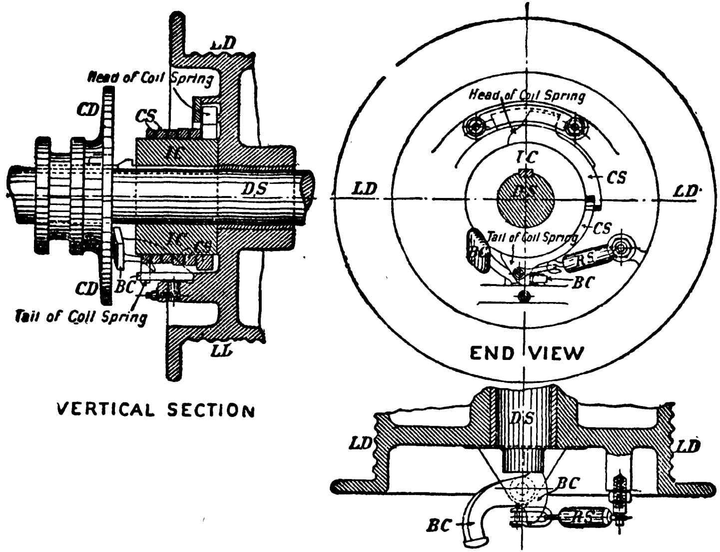
Switchboard Instruments.—"A lamp, an ammeter, a voltmeter, and an automatic circuit breaker (to prevent danger from overloading) to be provided and conveniently arranged on the switchboard, with all necessary connections to the main conductors."

Conductors and Insulation.—"The conductors to be of suitable standard sizes, corresponding to the Cable Makers' Association List. The conductors are to be connected from the terminals of the motors to the controlling switches, from thence to the cut-outs, and therefrom to suitable connections passing through the centre post to sockets at the bottom of crane carriage, and there provided with plug connections to the twin cable. The conductors in the house to be provided with circular sliding contacts and collectors, so as to admit of the crane being slewed in either direction to any extent without difficulty. A portable pair of conductors, with plugs sweated on ends, to be provided to fit sockets on carriage, and also the wall-plugs along the quay. The twin conductor to be of such a length as to extend to a distance of 40 feet on either side of the centre of the crane. The insulation resistance to be not less than 3,000 megohms per mile for the larger, and 4,000 for the smaller conductors."

Wall Plugs.—"Wall-plugs to be fitted in the hydraulic supply hydrant-boxes, or elsewhere, along the quay, with suitable terminals for connection to the electric current supply mains. The boxes to be thoroughly water-tight, and every precaution to be taken against damp."

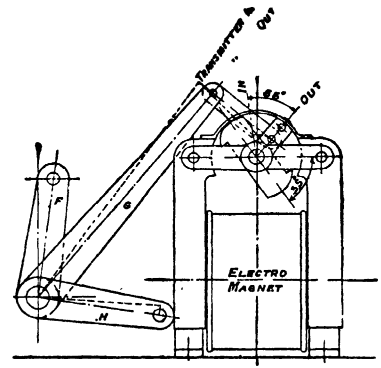
House.—"A substantially-built timber house to be constructed of sufficient size to give easy access to all parts of the machinery, and ample room for working the crane. A suitable electric radiator, or stove, to be provided, and fitted with a graduated resistance switch, to allow of varying degrees of heat being obtained. Two 32 candle-power Ediswan glow lamps, and two plugs for portable lamps, to be fitted up in house, and a cluster of glow lamps half way up the jib."

Details of Hoisting Brake and Levers for Working the 3-Ton Electric Crane.—In order to make the construction and action



Details of the Coil Clutch for 3-Ton Electric Crane.

LD	for	Lifting drum (*see previous and following views*).
DS	,,	Drum shaft.
IC	,,	Iron chilled cylinder.
CS	for	Coil spring of soft iron.
BC	,,	Bell crank lever.
RS	,,	Releasing spring for BC.
CD	,,	Clutch friction disc.

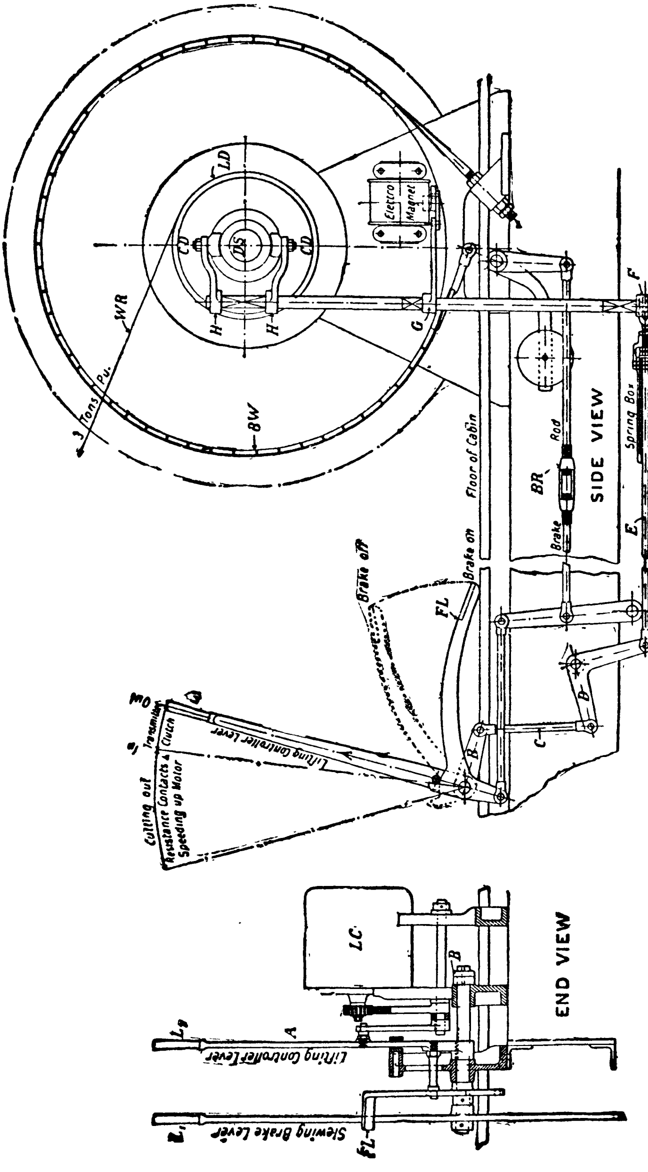


Electro-Magnet and Levers for Working Coil Clutch.

Index to Parts of Figs.

On opposite page and the above one.

L_1	for	Lever to slewing brake.
L_2	,,	Lever to L C and C D.
L C	,,	Lifting motor controller.
A to H	,,	Levers from L_1 to C D.
D S	,,	Drum shaft.
L D	,,	Lifting drum.
W R	,,	Wire rope.
B W	,,	Brake wheel.
B R	,,	Brake rod.
F L	,,	Foot-brake lever to B R.



LEVERS AND ELECTRO-MAGNET FOR WORKING STOTHERT & PITT'S 3-TON ELECTRIC CRANE.

of these details clear to the student, special views have been herewith reproduced from drawings kindly supplied by the makers.

Coil Clutch.—In the accompanying figures this has been shown by a vertical section, end view, and sectional plan. It will be understood, after comparing the index to parts with these views, that when the clutch friction disc, C D, is pressed forward by the levers, H (shown on the full-page views), it pushes the end of the bell crank lever, B C, so that its inner end pulls upon the tail of the coil spring, C S. This action causes C S to wind close and grip hard upon the iron chilled cylinder, I C. Consequently, we have the lifting drum, L D (which is directly connected to the inner end or head of the coil spring, C S), thereby coupled, through C S, to I C and D S, the drum shaft. The other end of this shaft is keyed to L W, which gears with the lifting motor pinion, L P, and hence with L M, the lifting motor. The load can then be lifted to the desired height.

Braking.—When the clutch friction disc, C D, is disengaged from B C, the latter is brought back to its normal position by the releasing spring, R S. This relieves C S from contact with I C. The lifting drum, L D, with its brake wheel, B W, are thus instantly disconnected from the motor. The load is then, however, held up in position by the brake-strap due to the driver pressing down the treadle or foot lever, F L. The load can now be lowered at any desired speed by simply easing the foot lever, F L, which acts directly on the brake rod, B R, and brake strap of the brake wheel, B W, as shown by the side view on the full-page set of figures. It will thus be seen, that the brake is normally "off," as in a steam crane. The driver lowers a load by pressing his foot more or less on the treadle, thus tightening or slackening the brake-strap in precisely the same way as he would do with a steam crane. The makers admit, that this method of actuating the brake may at first sight appear to be not so automatic or ideal a plan, as that of permitting the weight attached to the brake lever to keep the brake-strap tight on its brake wheel; but, they find it to be much better in actual practice. It will be observed, that if the electric current failed when a load had been raised, and if the driver then stood still and did nothing, the load would undoubtedly drop! However, the current has not been found to fail under these circumstances, and the driver has ample time to press the treadle and check the load.

Lifting.—It will be observed that the first part of the motion of lever L_2 to the left (*i.e.*, from the words "out" to "in"),

puts the clutch, C D, into action, and at the same time it brings the transmitter switch of the lifting controller, L C, into circuit with the lifting motor, L M. The remainder of the movement of lever L_2 in the same direction (*i.e.*, to the left) speeds up the motor, L M, by cutting out resistances in the controller of the motor circuit.

Lowering.—On the other hand, the action of putting on or down the treadle foot-brake lever, F L, for the purpose of stopping the motor before lowering a load, automatically throws back the clutch, C D, and disconnects the motor electrically from its controller. The lifting drum, L D, with its attached brake wheel, B W, are now free to move quite independently of the lifting motor and the drum shaft. The load can then be lowered at the desired speed by the required pressure on the foot lever.

Action of the Electro-Magnet.—In the Clyde Trust 3-ton electric crane, Messrs. Stothert & Pitt have adopted an alternative patent arrangement of electro-magnet for engaging and disengaging the clutch friction disc, C D. When it is used, the levers B, C, D, E, F, and spring box (as shown by the side view) are removed. A separate plan of this electro-magnet, with its levers, G and H, are shown. It will be understood, that the current, in passing from the controller, L C, to the lifting motor, L M, traverses and energises the coils of this electro-magnet.

Abstract of Report, Table and Curves on Comparative Trials for Efficiency of the previously-described 3-Ton Hydraulic and Electric Cranes.*—The electric crane was erected on the South Quay, Prince's Dock, and started to work on the 24th March, 1902. It has worked since then with the most satisfactory results, and there has been no hitch or trouble of any kind with the electrical machinery. From that date up to the 28th February, 1903, this crane discharged and loaded minerals and general cargo amounting to 43,954 tons, as compared with 40,833 tons dealt with by the 3-ton hydraulic crane alongside of it, whilst working under precisely similar conditions. Both cranes were tested on the 19th and 20th March, 1903, and the results given in tabulated form, indicate that the efficiency of the electric is greater than that of the hydraulic crane at full load, both in working ore and general cargo. The efficiency of the electric crane is also very much greater at the lighter loads, while the cost of producing and distributing electric power to the cranes is less than that of hydraulic power. The curves of performance of these cranes under tests are shown on diagrams 1 and 2. By comparing the corresponding curves, C, on these two diagrams—which show the cost of power used in discharging 1,000 tons of general cargo at varying loads—the difference in favour of the electric crane may be appreciated. For example, in working with loads of

1 ton	2 tons	2½ tons	3 tons

the cost of power used by the hydraulic crane in discharging 1,000 tons of general cargo is

48/3	25/5	20/6	17/2

and by the electric crane

15/11	16/2	16/1½	16/6½

It will be observed, that the costs for the hydraulic crane are nearly in inverse proportion to the loads raised, for the reason that in this type of crane, the same foot-pounds of water are used to lift the empty hook as to lift the maximum load. On the other hand, with the electric crane the cost is practically the same at all loads. This is due to the valuable feature, that the electrical power used, is much more nearly proportional to the power required to raise the load, than is the case with hydraulic cranes. By raising the electrical pressure at the central power-house from 260 volts (its present value) to, say, 500 or 600 volts

* I am indebted to Mr. George H. Baxter, M.I.E.S., for the liberty to make this abstract from his report to the Clyde Trustees.

(as contemplated for the new Clydebank Dock), and by working upon a greatly increased scale, it is probable that the present average cost of 2·558 pence per Unit will be further reduced.*

Relative Cost of Hydraulic and Electric Power.—Since the hydraulic and electric are the only two practicable and available systems of power transmission for dock appliances, such as cranes, capstans, and coal hoists; and, since both systems are equally reliable and convenient, the choice of one or the other depends entirely on which is the more economical, not only in regard to the cost of production and distribution, but also as regards the cost of utilisation.

One system may cost considerably less to produce and distribute power, but in utilisation the gain in point of economy may not be in favour of that system

There are, indeed, so many factors to be taken into account that, for purposes of comparison, and also to determine whether it would cost less or more to purchase or produce power, the cost of each system should be ascertained at three distinct stages, viz. :—(1) The cost of purchase or production. (2) The cost of transmission or distribution. (3) The cost of utilisation.

In order to compare the costs of hydraulic with that of electric power, it is necessary to reduce both systems to a common standard, and to determine in each case the cost of *one horse-power hour* delivered at the appliance where the power is to be utilised. This is the most convenient unit to adopt, and is applicable alike to both systems.

In making up the working costs, the capital expenditure and working costs have been taken from the annual accounts, and embrace all the capital expended, up to the end of 1902, on buildings, foundations, culvert, machinery, and plant for each of the two systems.

The outputs of hydraulic power during the periods under notice, have been found by keeping strictly accurate records of every revolution of the pumping engines. The outputs of electric power have been got from carefully kept daily records of the hours of lighting, pressure, and current used.

The conclusions arrived at are, that electric power costs less to produce, distribute, and utilise than hydraulic power.

Crane Tests.—Tests were conducted under favourable conditions in order to ascertain the following :—

1. The rapidity of working of both types.
2. Their efficiencies.
3. The cost of power used in doing the same amount of work,

* In a large well-regulated power-house the Cost of Electric Power may be as small as 1d. per Board of Trade Unit—*i.e.*, 1d. per 1,000 volt-ampere-hours or watt-hours.

viz., to raise 30 feet, slew 100 degrees, and discharge into waggons 1,000 tons of cargo and 1,000 tons of ore, when working at loads varying from 1 to 3 tons for cargo, and at $2\frac{1}{2}$ and 3 tons for ore.

Particulars of Hydraulic Crane and its Gauges.—The pressure in the hydraulic 7-inch main was maintained at 750 lbs. per square inch at the Power-House. The 3-ton hoisting ram was used. It is 11 inches diameter, and displaces ·66 cubic foot of water per foot of travel, or ·11 cubic foot per foot of hoist, the velocity ratio being 1 to 6. The turning ram is 9 inches diameter, travels $15\frac{1}{2}$ inches, and uses ·57 cubic foot in slewing through 100 degrees. The water used was measured by the displacement of the rams, and 5 per cent. was allowed for slip and leakage.

Pressure gauges were fitted on the main pressure pipe at the hydrant connected to the crane, and to the upper parts of the hoisting and slewing cylinders. The pressures as indicated by these gauges were noted at each lift.

Particulars of Electric Crane and its Meters.—The hoisting gear is single reduction, and is worked by a 50 H.P. Siemens' motor at a pressure of about 240 volts. The pressure at the Power-House was maintained at 260 volts, whilst the current naturally varied with the load. The turning motor is 10 H.P.

The electric crane was equipped with Siemens' volt and ampere meters, and also with Chamberlain & Hookham's watt meters, which were overhauled and found to be accurate before the tests were commenced.

Details of Testing Operations.—In testing for rapidity of working, each crane was provided with two tipping buckets, one for the load and one for the empty bucket. For convenience, the tests were carried out to specific instructions in double cycles, by working the cranes as rapidly as possible for two hours, under each separate load of 1 ton, 2 tons, $2\frac{1}{2}$ tons, and 3 tons, in such a way, as to imitate as nearly as possible actual conditions of working, without being subjected to the delays so frequently experienced in discharging ore or general cargoes.*

* In calculating the weight of cargo discharged per hour, allowance was made for the 9-cwt. empty buckets and the 2-cwt. hoisting chain balls, as follows; $(9 - 2)$ cwts. = ·35 ton. Now, this weight was lifted only 10 feet during the return journey with the empty bucket, and = $(·35 \times 10)$ ft.-tons of work. But each ton of paying load is lifted 30 feet = 30 ft.-tons of work. Hence, for every 30 ft.-tons of general cargo work, there would have to be added in this test, 3·5 ft.-tons for the empty bucket minus the chain ball. Or, $\left(·35 \times \frac{10}{30}\right) =$ ·116 ton, as shown in the two following tables.

Curve Diagrams 1 and 2.—The principal curves to which attention may be directed are the speed lines, S, which show that both cranes are nearly alike as regards speed at all loads—varying from 260 to 165 feet per minute in the case of the hydraulic crane, and from 230 to 165 feet per minute in the electric crane, at loads of ½ and 3½ tons respectively.

The efficiency line, E, for the hydraulic crane rises in a straight line from 9½ per cent. to 66 per cent. for loads of ½ and 3½ tons respectively. The efficiency line, E, of the electric crane rises in a full, well-rounded curve, showing at

½ ton	1 ton	2 tons	2½ tons	3 tons	3½ tons

efficiencies of

31%	49·34%	73·14%	77·1%	75·3%	70%

The other lines, C, show the cost of power used in raising 30 feet, slewing 100 degrees, and discharging into waggons 1,000 tons of cargo. It will be seen, that the cost line for the hydraulic crane rises in a steep curve at the light loads, while that for the electric crane is a straight line nearly parallel with the abscissæ, showing how the cost increases in the case of the former when working at light loads, while the cost is practically constant at any load with the electric crane.*

* I have added to these two diagrams E_C lines. In the case of the hydraulic crane this dotted curve, E_C, indicates the percentage efficiency as found by the pressure gauges attached directly to the hydraulic cylinder. This curve, therefore, represents the ratio of the power got out at the crane hook (when lifting loads 30 feet in height), to the power put into the cylinder during that time. It clearly shows the effect of the loss of pressure between the hydrant on the main supply pipe and the ram cylinder. It also shows by comparison the efficiency of the plant between the crane hook and the cylinder ram with that between the same hook and the power put into the mains at 750 lbs. per square inch.

In the case of the electric crane, the dotted curve, E_C, is derived from special sets of ten observations for each load, taken on March 20th, 1903. The times of simply lifting loads 30 feet were accurately noted, as well as the readings on the crane volt and ampere meters during these respective lifts, quite independently of the meters at the power-house. This curve, therefore, represents the best net combined electrical and mechanical efficiency of the crane, whilst merely lifting different loads. It indicates the ratio of the power got out at the crane hook, to the power put into the lifting motor, or the crane efficiency

Note.—Students may refer here to *Engineering* of June 19 and July 3, 1903, for reports upon two papers, "Economical Speeds of Cranes," and "Hydraulic *versus* Electric Cranes for Docks," read at the London Engineering Conference.

TABLE I.—RESULTS OF TESTS OF 3-TON HYDRAULIC CRANE WORKED AS IF DISCHARGING GENERAL CARGO.

LOADS (W).	1 Ton.	2 Tons.	$2\frac{1}{2}$ Tons.	3 Tons.
Duration of trial, hours,	2	2	2	2
Pressure in mains, lbs. per sq. in.,	750	750	750	750
Pressure on ram—				
Hoisting, . . ,,	240	400	480	560
Lowering, . . ,,	140	300	380	460
Slewing to windward, ,,	500	500	500	500
,, leeward, ,,	350	350	350	350
1. NUMBER OF CYCLES PER HOUR, ...	56	54	52	38
Time of lifting W 30 feet (T), . . seconds,	7·5	8·57	9·25	10
Speed of hoisting loads $(S) = \frac{30 \times 60}{T}$ ft. per min.,	240	210	194·6	180
H.P. taken out of crane $(L.H.P.) = W \times S \div 33{,}000$,	16·29	28·51	33	36·65
Horse-power, by water $(W.H.P.) = \frac{95 \times 750 \times S}{33{,}000 \times 6}$,	86·36	75·56	70	64·77
2. MECHANICAL EFFICIENCY (*Hoisting only*), $\% = 100 \times \frac{L.H.P.}{W.H.P.}$, .	18·87	37·73	47·14	56·6
Cargo discharged per hour (allowing for empty bucket $·35 \times \frac{10}{30}$ ton), tons,	(1·116 × 56) 62·5	(2·116 × 54) 114	(2·616 × 52) 136	(3 116 × 38) 118·5
Water used per cycle—				
Hoisting 30 feet, cub. ft.,	3·3	...	...	...
Slewing 100°, ,,	·57	...	...	...
Hoisting 10 feet, ,,	1·1	...	...	...
Slewing 100°, ,,	·57	...	...	...
Slip, 5 per cent., ,,	·28	..	...	...
Total cubic feet, .	5·82	5·82	5·82	5·82
Water used per hour, cub. ft.,	326·5	314·5	303	221·5
Cargo discharged per hour, tons,	62·5	114	136	118·5
Water used per 1,000 tons cargo, . . cub. ft.,	5,224	2,759	2,228	1,869
3. COST OF HYDRAULIC POWER used in discharging 1,000 tons general cargo, at 110·45 pence per 1,000 cubic feet of water at 750 lbs. per square inch (C), . .	48/	25/5	20/6	17/2

TABLE II.—RESULTS OF TESTS OF 3-TON ELECTRIC CRANE WORKED AS IF DISCHARGING GENERAL CARGO.

LOADS (W)	1 Ton.	2 Tons.	$2\frac{1}{2}$ Tons.	3 Tons.
Duration of trial, hours,	2	2	2	$\frac{1}{4}$
E.M.F. at motor terminals, . . volts,	240	245	240	245
Current when hoisting loads, amps.,	95	110	125	140
Current when lifting empty bucket, . amps.,	60	60	60	60
Current slewing to windward, amps.,	15	35	35	40
Current slewing to leeward, . . amps.,	15	15	15	15
1. NUMBER OF CYCLES PER HOUR, . . .	60	53	48	43
Work done by hoisting W 30 feet, . . foot-lbs.,	67,200	134,400	168,000	201,600
Time of lifting W 30 feet (T), . seconds,	8·1	9·25	9·85	10·6
Speed of hoisting loads, $S = \frac{30 \times 60}{T}$ ft. per min.,	222·2	194·[illegible]	182·7	170
H.P. taken out of crane $(L.H.P.) = W \times S \div 33{,}000$,	15·08	26·42	31	34·62
Horse-power expended, $(E.H.P.) = \frac{\text{volts} \times \text{amps.}}{746}$,	30·56	36·12	40·21	46
2. COMBINED MECHANICAL AND ELECTRICAL EFFICIENCY, $°/_{\circ} = 100 \times \frac{L.H.P.}{E.H.P.}$ (*Hoisting only*), . .	49·34	73·14	77·1	75·3
Cargo discharged per hour (allowing for empty bucket ·35 × $\frac{10}{30}$ ton), tons,	(1·116 × 60) 67	(2·116 × 53) 112	(2·616 × 48) 125·5	(3·116 × 43) 134
Units used per hour at various loads, . . .	5	8·5	9·5	10·4
Units required for 1,000 tons cargo (U), . .	74·6	75.8	75·7	77·6
3. COST OF ELECTRIC POWER used in discharging 1,000 tons general cargo, at 2·558 pence per unit (C),	15/11	16/2	16/1½	16/6½

DIAGRAM No. 1.

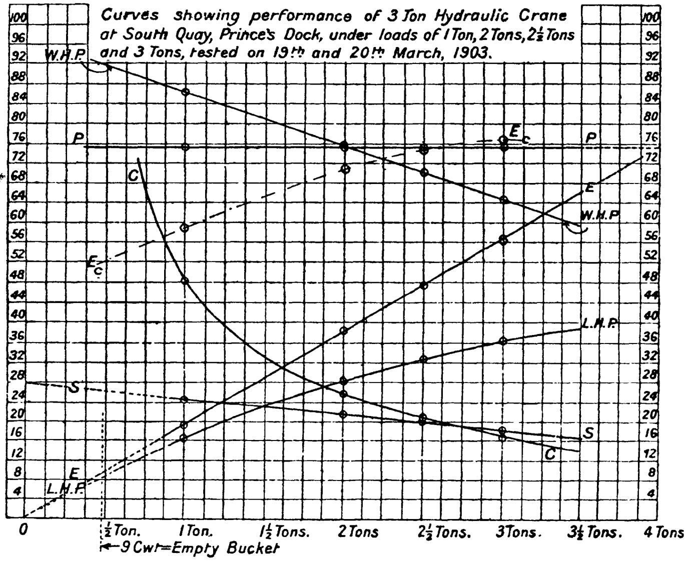
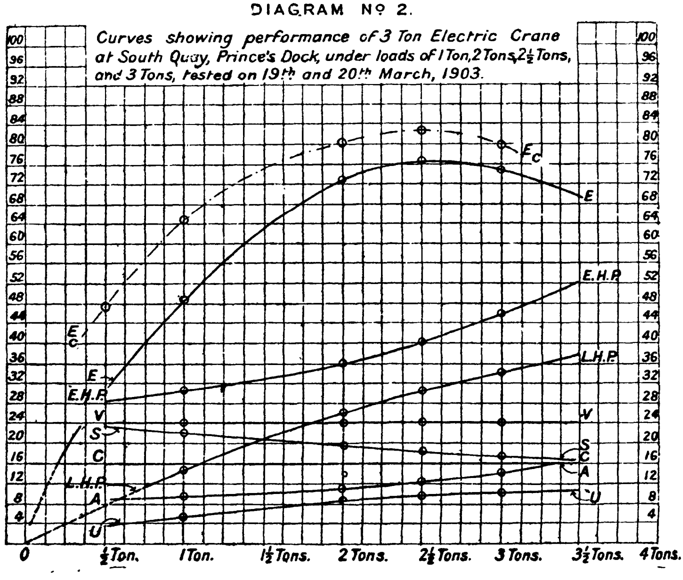


DIAGRAM No. 2.



EXPLANATION OF EFFICIENCY CURVES.

HYDRAULIC CRANE.

P. Pressure in mains during tests.
Each division = 40 lbs. per square inch.

S. Speed of hoisting in feet per minute.
Each division = 40 feet per minute.

W.H.P. Horse-power put into crane by water at 750 lbs. per square inch, hoisting only.
Each division = 4 H.P.

L.H.P. Horse-power taken out of crane in hoisting only.

E. °/$_\circ$ Mechanical efficiency $= \frac{\text{L.H.P.}}{\text{W.H.P}} \times 100$ (*hoisting only*).
Each division = 4 per cent.

E_C. °/$_\circ$ Efficiency $= \frac{\text{Power got out at crane hook}}{\text{Power put into hydraulic ram}}$, when *merely* lifting different loads 30 feet during special trials.
Each division = 4 per cent.

C. Cost of power used in raising 30 feet, slewing 100°, and discharging into waggons 1,000 tons of cargo.
Each division = 4 shillings.

ELECTRIC CRANE.

V. Volts while hoisting at speed (**S**).
A. Amperes ,, ,,
Each division = 40 volts and also = 40 amperes.

S. Speed of hoisting in feet per minute.
Each division = 40 feet per minute.

E.H.P. Horse-power delivered to hoisting motor.
Each division = 4 H P.

L.H.P. Horse-power taken out of crane in hoisting only.

E. °/$_\circ$ Electrical and mechanical efficiency $= \frac{\text{L.H.P.}}{\text{E.H.P.}} \times 100.$
Each division = 4 per cent (*when hoisting only*).

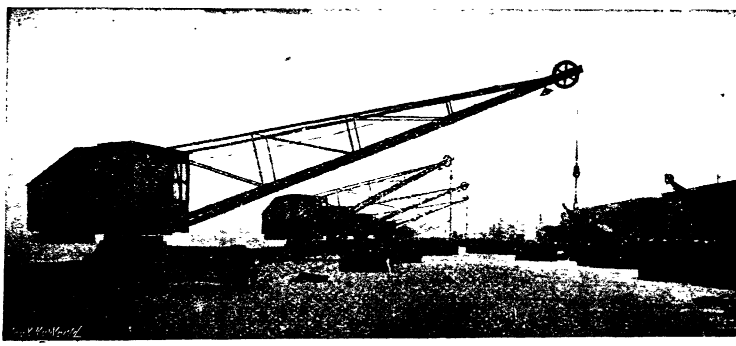
E_C. °/$_\circ$ Crane efficiency $= \frac{\text{Power got out at crane hook}}{\text{Power put into motor}}$ when *merely* lifting different loads 30 feet during special trials.
Each division = 4 per cent.

C. Cost of power used in raising 30 feet, slewing 100°, and discharging into waggons 1,000 tons of cargo.
Each division = 4 shillings.

U. Units used per hour at various loads.
Each division = 4 units.

Electric Cranes for Manchester Ship Canal.—As one more instance of the rapid progress which the application of electricity to cranes is making, we here show two views of those recently made and fitted by George Russell & Co., of Motherwell, for the Manchester Ship Canal Warehouse Co. There are twenty-two cranes placed on the roof of the warehouse. Of these, twelve are portable cranes on the dock side of the warehouse for lifting 30 cwts, with a radius of 42 feet. The remaining ten are fixed cranes on the land side of the warehouse, three of which lift 30 cwts., and seven 10 cwts., both having a radius of 16 feet.

The electrical motors and controllers were supplied by the British Thomson Houston Co., Rugby. The hoisting motors for

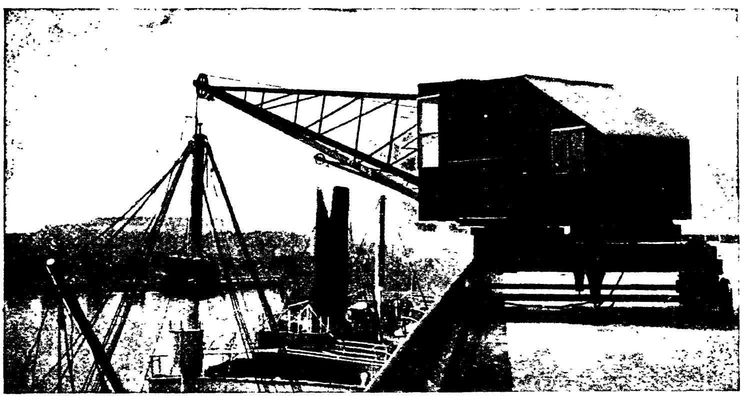


Fixed and Movable Electric Cranes on the Manchester Canal Warehouse.
(Made by George Russell & Co., Ltd., Motherwell, N.B.)

the 30-cwt. cranes are rated at 28 H.P. when their armatures have a speed of 355 revs. per minute, and are supplied at 500 volts. They are geared direct to the lifting barrels by single reduction machine-cut spur wheels. The barrel of each of these cranes is 24-inches diameter, and the lift is effected by a single wire rope. The speed of lifting was specified to be 200 feet per minute, with a full lift of 75 feet. A strap brake operates upon a brake wheel, keyed upon the barrel shaft. Under normal conditions the brake bears upon its brake wheel through the action of a lever weight, just as in the previously described Clyde Trust Electric Crane. But, in this instance, the weight is raised by an electro-magnet in series with the lifting motor,

so that whenever current is switched on to the motor the weight is lifted and the brake released. In the event of the current failing through any cause the pull of the magnet ceases, and the weight automatically applies its brake to prevent the load from falling. The brake may be also released by a foot tread, so that the load may then be lowered without using any current.*

From a final test of one of the portable cranes it was found that whilst lifting four bales of cotton at a time, having an



MOVABLE ELECTRIC CRANE UNLOADING A STEAMER AT THE MANCHESTER CANAL.

average weight of 3,136 lbs., through 43½ feet, that the time was 11·5 seconds (or 227 feet per minute), current 38·5 amperes at 510 volts; consequently, the efficiency under these circumstances reached 82 per cent.

* For a fuller description of the details of these cranes students may refer to the March and April numbers of the leading British electrical and mechanical journals, such as *The Tramway and Railway World*, April, 1903, to which paper I am indebted for the two views here reproduced.

LECTURE II.—QUESTIONS.

1. Explain why the force necessary to drive a machine does not vary in exact proportion with the load.

2. With a pair of three-sheaved blocks it is found by experiment that a weight of 40 lbs. can be raised by a force of 10 lbs., and a weight of 200 lbs. by a force of 40 lbs. Find the general relation between P and W, and also the efficiency when raising 100 lbs. *Ans.* $P = \frac{3}{16} W + \frac{5}{2}$; ·784 or 78·4 per cent.

3. With a screw jack it is found that a force of 47·5 lbs. must be applied at the end of the handle to lift 1·5 tons, and a force of 85 lbs. to lift 3 tons. Find what force will be required to raise 2 tons. *Ans.* 60 lbs.

4. If the length of the handle in the above example be 2 feet and the pitch of the screw $\frac{3}{4}$ inch, find the efficiency in each case. *Ans.* 35·3, 39·5, and 37·3 per cent.

5. Find the "*friction pressure*" of a steam engine which requires a mean effective pressure of 24 lbs. per square inch to drive it at full load, 20 lbs. being taken up in overcoming the load. At three-quarters load a mean effective pressure of 18·5 lbs. is required, of which 15 lbs. is similarly taken up. *Ans.* 2 lbs. per square inch.

6. It is found that a force of 2 lbs. must be applied to the handle of a crane in order to wind up the rope when no weight is attached and of 80 lbs. when lifting a weight of 10 cwts. If the velocity ratio be 20, find the efficiency in this last case and also when the force at the handles is lessened so as just to allow the weight to descend. *Ans.* 70 per cent.; 57 per cent.

7. A machine is concealed from sight except that there are two vertical ropes; when one of these is pulled downwards the other rises. If the falling of a weight, A, on one, causes a weight, B, on the other to be steadily lifted, first when A is 100 lbs. and B is 3,250 lbs., second when A is 50 lbs. and B is 1,170 lbs., what is the probable value of A when B is 2,000 lbs.? How would you find the efficiency of this lifting machine in these three cases? What new measurement must be made?

Ans. 69·95 lbs.

8. A machine is concealed from sight, except that there are two vertical ropes; when one of these is pulled downwards the other rises. If the falling of a weight, A, on one, causes a weight, B, on the other to be steadily lifted, first when A is 12 lbs. and B is 700 lbs., second when A is 7·6 lbs. and B is 300 lbs. What is A likely to be when B is 520 lbs.? If B rises 1 inch when A falls 70 inches: what is the efficiency of this lifting machine in each of the three cases?

9. In a lifting machine an effort of 26·6 lbs. just raises a load of 2,260 lbs., what is the mechanical advantage? If the efficiency is 0·755, what is the velocity ratio? If on this same machine an effort of 11·8 lbs. raises a load of 580 lbs., what is now the efficiency?

10. If the diameter of the 3-ton lifting ram of the hydraulic crane described in this lecture is 11 inches, and its stroke be 5 feet when the chain hook rises 30 feet, and if the inside diameter of the water service pipe from the power-house to the crane dock hydrant opposite this crane is 7 inches; find what length and weight of a column of water in the service pipe is used at each complete 5-foot stroke of the lifting ram.

11. Referring to the previous question, if a 3-ton load be lifted 30 feet high in 10 seconds, and if the lengths of 7-inch service piping between the crane hydrants and the power-house accumulator be 500 and 1,000 feet respectively for two different cranes, what will be the loss in pressure in lbs. per square inch between the power-house and each of these two hydrants, when accumulator applies a steady pressure of 750 lbs. per square inch at the power-house, and if the coefficient of friction between the water and its service pipe (or the "friction factor") is ·02 at the velocity under these circumstances? Also, calculate (*a*) the ft.-lbs. of work; (*b*) the watt-hours of work lost through the friction in the pipe at each lift of 3 tons; (*c*) the ft.-lbs. per second; (*d*) the watts or the lost power in the water service pipe during the lift.

12. Referring to Table I. of tests and the curves for the 3-ton hydraulic crane in this Lecture, how do you account for the differences in the loss of the pressures per square inch between the water in the mains and the pressures on the ram whilst hoisting different loads? What would be the calculated corresponding loss of pressure in each case between the power-house and the crane hydrant if the distance of the main supply pipe was 1,000 feet long and the times of lifting 1, 2, 2½, and 3 tons as stated in Table I.?

13. Check by calculations the several values for W.H.P., L.H.P., E.H.P., E., and C., in the case of the hydraulic and electric cranes described in this Lecture, and draw their curves upon squared paper neatly to a scale at least double the size of that given in the text.

14. Referring to Tables I. and II., &c., in this Lecture, calculate and compare the loss of power in ft.-lbs. per second (or in watts) for the hydraulic and the electric mains whilst lifting 1, 2, 2½, and 3 tons, and show clearly which system of distribution of power is more efficient. How would you propose to render the distribution of power from the power-house to the electric crane more economical?

Lecture II.—A.M.Inst.C.E. Exam. Questions.

1. What is meant by the efficiency of a machine? How would you measure the efficiency (1) of a screw-jack, (2) of a steam pump, (3) of a turbine?

2. An hydraulic company charges 15d. per 1,000 gallons of water at 820 lbs. per square inch; what is the cost per H.P. hour?

3. The saddle of a lathe weighs 5 cwts., and it is moved along the lathe bed by a rack and pinion arrangement. What force applied at the end of a handle 10 inches long will be just capable of moving the saddle, supposing the pinion to have twelve teeth of 1¼ inch pitch, the coefficient of friction between the saddle and lathe bed to be 0·13, and if 35 per cent. of the work done in working the rack and pinion is expended in overcoming friction.

4. An experiment on a windlass to determine the effort required to lift different loads, gave the following results:—

Load (lbs.)	0	28	56	84	112	140	168	196	224
Effort (lbs.)	1·5	2·75	4	5·2	6·4	7·6	8·8	9·9	11

The velocity ratio of the machine was 52. Plot the results on squared paper, showing the effort curve on a load base, and draw also the friction and efficiency curves, finding at least five points in each. Find the effort for a load of 210 lbs., and find the mechanical advantage and mechanical efficiency of this load.

LECTURE III.

BERNOUILLI'S THEOREM.

CONTENTS.—Energy of Flowing Water—Bernouilli's Theorem—Jet Pumps, Injectors, and Ejectors—Hydraulic Ram—Example—Venturi Law and Meter—Theory of the Venturi Tube—Effect of Varying the Throat Ratio—Uses of the Venturi Meter—Kent's Recorder—Friction Curve—Questions.

WE have considered, up to the present, problems due to the pressure of fluids at rest—or hydrostatics, as it is called; we come now to consider hydrokinetics—*i.e.*, problems involving the flow of water.

Energy of Flowing Water—Bernouilli's Theorem.—When a liquid is flowing in a pipe or channel, it possesses kinetic energy in virtue of its motion in addition to the potential energy due to its position and pressure; and the total energy is the sum of these three.

Let v = Velocity of the liquid.
„ h_1 = Length of its pressure column.
„ h_2 = Height above the datum level.
„ H = Height of the free surface of the still water above the datum level.
„ m = Mass of the portion of the liquid under consideration.
„ g = Acceleration due to gravity.
„ f = Frictional loss of head.

Then,

The energy of pressure $= m\,g\,h_1$
„ of position $= m\,g\,h_2$

And, „ of motion $= \frac{1}{2}\,m\,v^2.$

Hence, **The total energy** $= m\,g\left(h_1 + h_2 + \frac{v^2}{2\,g}\right).$ (I)

Or, **Energy per unit mass** $= g\left(h_1 + h_2 + \frac{v^2}{2\,g}\right)$. (I_a)

It follows from the principle of the *Conservation of Energy* that so long as no work is spent on friction, this total remains constant whilst the water flows along the pipe or channel. Therefore, for a frictionless liquid in which there are no eddies:—

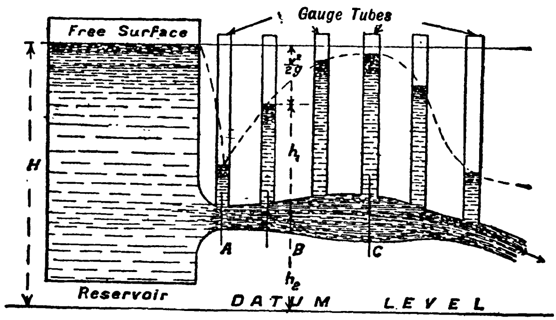
$$h_1 + h_2 + \frac{v^2}{2\,g} = \text{a constant} = \mathrm{H}. \quad . \quad . \quad . \quad \text{(II)}$$

If, p = Pressure per unit area at any point in the liquid,
And, w = Weight of unit volume of the liquid,

Then, $h_1 = \frac{p}{w}$;

Hence, $\frac{p}{w} + h_2 + \frac{v^2}{2g}$ = a constant = H. . . . . . . (III)

This equation is known as *Bernouilli's Theorem.*



PRESSURES IN A FRICTIONLESS PIPE.

If a certain amount of energy mgf be absorbed by friction between some vertical datum section such as A and the section under consideration which may be at B or C.

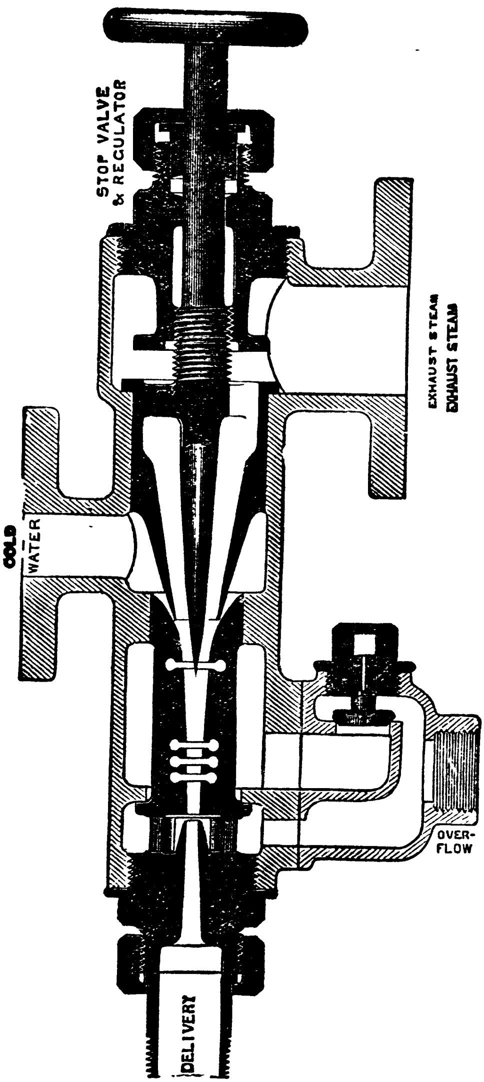
Then, $\frac{p}{w} + h_2 + \frac{v^2}{2g} + f$ = a constant = H. . . (IV)

Let a = The cross area of the pipe at any section.
„ v = „ velocity of the water at the same section.
„ Q = „ quantity or volume of water passing every section in unit time. This volume is constant for all sections during a steady flow of the liquid.

Then, $av = Q$; or, $v = \frac{Q}{a}$. . . . . . . (V)

This shows, that the velocity is great when the cross section of the pipe is small, and *vice versâ.*

In the figure, we have shown small vertical gauge tubes placed at intervals A, B, C, &c., along the pipe, in order to indicate the pressure of water at these points by the height to which it rises in them. It will be observed that where the pipe is level or nearly so, the pressure is greatest where its cross section is largest and consequently the velocity is least. This follows directly from



Exhaust Steam Ejector.

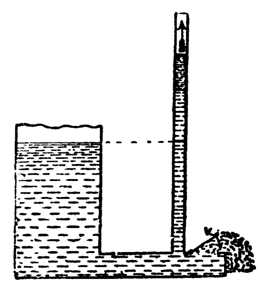
Bernouilli's theorem, since the height h_2 above the datum level is practically constant.

Jet Pumps, Injectors, and Ejectors. — From the previous illustration and explanation of the alteration in the pressure of water as it flows through a pipe of varying size, it will be readily understood that if a stream of water be rapidly forced through a tapered passage, its pressure may be so lowered below the surrounding atmosphere, that it can draw in more water from another source connected to the smaller end of the taper. If the whole of the water be then conducted along a gradually enlarged passage, its pressure will increase and the outflow can take place at a higher level than the intake of the induced stream, but lower than the free surface of the driving water. The late

Prof. James Thomson, of Glasgow University, designed his jet pump upon this principle, and Prof. Bunsen used a jet of water to produce a vacuum in his air pump. Steam jets, compressed air, and water under pressure, have frequently been used to create a blast of air, to feed petroleum into furnaces, to produce a sand blast for engraving or cleaning purposes, and to transfer granular materials from one position to another.

Injectors for feeding steam boilers with water also work upon this principle; but since they are greatly assisted by the condensation of the steam as it comes into intimate contact with the suction water, the latter can be forced into the same boiler or another vessel having the same pressure as, or even a higher pressure than, that which supplies the injector with steam.* Exhaust steam ejectors also depend upon the above action, and are sometimes used to replace both the ordinary jet condenser and air pump in connection with condensing engines. As will be seen from the accompanying figure, when the regulating stop valve is screwed upwards by turning the hand-wheel, the exhaust steam from the engine cylinder enters by the right-hand upper pipe and mixes with cold water coming through the left-hand one, thereby becoming condensed and producing the desired vacuum. The combined condensed steam and condensing water then flow from the delivery pipe into the hot well, whilst any throttled discharge, or some of the live steam that may have been used for blowing through and starting the ejector, can escape into the same place by the overflow pipe. As we shall prove further on, apparatus of this kind cannot have a very high efficiency since the mixing up of quick and slow moving streams results in a considerable loss of mechanical energy.

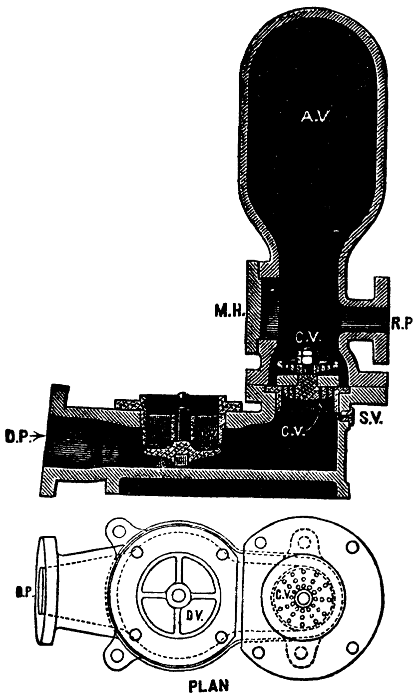
Hydraulic Ram.—This apparatus was invented about 100 years ago by a Frenchman named Montgolfier. It is one of the simplest, most durable, and efficient machines for raising water to a greater height than the source of supply. The energy stored up in water descending from a comparatively low elevation is utilised to raise part of the same water to a much higher level, of from three to thirty times the vertical height of the original fall. The principle upon which the apparatus works will be understood from



PRINCIPLE OF THE HYDRAULIC RAM.

* For a description of Giffard's and other steam injectors see the author's *Text-Book on Steam and Steam Engines*.

a consideration of the foregoing figure. If the valve V be held down firmly on its seat, and the left-hand vessel be filled with water to a certain height, it will rise to the same level in the right-hand open pipe. If the valve be now released for a short time, water will flow under the action of gravity along the horizontal passage and escape at the open valve with a velocity proportional to the square root of the "head" or vertical height of the free surface above the valve. On suddenly closing the valve, the kinetic energy of the moving water will be partly spent in raising the right-hand column to a greater height than the free surface of the water in the left-hand vessel. Now, if we introduce a check valve at the foot of the long column, so as to prevent this water from falling down again, and an air vessel to act as a cushion, we can repeat the operation continually, so as to produce a flow of water up the pipe.



HYDRAULIC RAM, BY THE GLENFIELD COMPANY, KILMARNOCK.

The machine, as made and supplied by the Glenfield Company of Kilmarnock, is illustrated by a vertical section and plan in the accompanying figure. Water flows from a cistern, tank, pond, or dam, through a cast- or wrought-iron pipe, technically called the drive pipe D P, to a hollow casting containing two valves. The first of these is named the escape or dash valve D V, which opens downwards, and the other the check valve C V, which opens upwards. Over the latter is fixed an air vessel A V, having a manhole door M H to the left and a delivery pipe, which is technically termed a rising pipe R P, to the right. If the

apparatus and all the pipes are duly connected to the supply and delivery tanks, and the dash valve D V be held up, until the water from the source of supply has filled not only the drive pipe D P, but also risen through the check valve C V and rising pipe R P to nearly the same level as the free surface in the supply tank, the whole will remain motionless or in a static condition. If we now depress the dash valve D V, and then let it go, the machine will immediately begin to work, and continue to work automatically without any attention or even oiling for years, until stopped by some accident or by the wearing out of one or both of the valves. Of course, the supply of water must be maintained, so that the drive pipe is always kept full. This pipe should not be throttled in any part, and the weight on the dash valve must be so carefully adjusted, that it will just overcome the internal pressure—*i.e.*, drop from its seat—and permit water to escape thereat. Then, the acceleration produced by gravity on the water coming down the drive pipe very soon produces a greater pressure on the dash valve than that due to the mere static pressure. This increased force suddenly raises it again to its seat, when the kinetic energy which has been imparted to the water lifts the check valve and forces some water into the air vessel. Whenever this kinetic energy has been spent, the compressed air in the air vessel, together with the weight of the check valve, causes it to close, and immediately thereafter the dash valve automatically opens. The same cycle of operations takes place over and over again, the air in the air vessel gets more and more compressed, and water rises higher and higher in the rising or delivery pipe, until it issues as a continuous stream from its mouth into the cistern or receiving tank. From this tank it may be drawn off at pleasure for all the various uses of a mansion-house or farm steading, &c.

The air vessel plays two important parts in each cycle of the operations of this interesting and useful apparatus. (1) The air contained therein acts as a cushion by minimising the water hammer action, which would otherwise stress the various parts, and tend to break the joints. (2) The air acts as a store of energy by taking up, during its compression, a part of the kinetic energy of the water, and then giving out the same gently, thus producing a constant flow of water through the delivery pipe. If the vertical height of the column of water in the rising pipe be about 34 feet above the check valve, the pressure per square inch on the upper surface of this valve will be one atmosphere, or, say, 15 lbs. on the square inch, and the air in the air vessel will be compressed to nearly that pressure,

and therefore occupy about half its original volume. If the column be 68 feet high in the delivery pipe, the pressure on the valve will be about 30 lbs. on the square inch, and the air in the air vessel will occupy one-third of its original volume, and so on. Hence, it is necessary to proportion the size of the air vessel to the vertical height through which the water has to be forced in the delivery pipe. Besides this, air becomes absorbed by water, and in a short time the air vessel, if small, would become entirely filled with water. The air vessel may, however, be kept charged with air in a very simple manner by the introduction of a snifter valve S V, screwed into the ram casing, immediately below the check valve. In its simplest, and probably its most efficient form, it consists of a brass plug with a very small hole drilled through its axis. Every time that water is forced through the check valve a very small quantity also passes through this tiny opening in the snifter valve; and each time that the check valve is forced down upon its seat a rebound or reaction of the water takes place, and produces a partial vacuum immediately underneath the check valve. Consequently, a little air is forced into this vacuum by the atmospheric pressure, and this air is carried up into the air vessel at the next stroke or pulsation, thus keeping up the necessary supply for effecting a continuous flow of water into the receiving tank. If everything about this machine is thoroughly tight and in good working order, and the valves are made of the best proportions and weights, an efficiency of from 80 to 90 per cent. can be obtained therefrom, and it has been found possible to work it with a minimum driving head of only three feet. The several causes for loss of efficiency are:—

(1) Eddies caused by the sudden stoppage of the water's motion.

(2) The friction of the water passing along the drive pipe D P, and the casing of the apparatus.

(3) The weight and friction of the dash valve D V, which has to be lifted at each stroke or pulsation.

(4) The weight and friction of the check valve C V, which has also to be lifted at each stroke.

(5) The slip of the check valve C V—*i.e.*, a slight quantity of water may slip back past this valve when in the act of closing.

(6) The friction of the water passing along the rising pipe R P.

(7) Any defects of tightness in the faces of the dash and check valves.

The sudden closing of the dash valve is only necessary to prevent the water spending a large portion of its energy in friction

at the restricted orifice while it is closing. Large rams are now made with special valves to stop the water gradually without throttling it, and so avoid the shocks caused by sudden closing without losing anything in efficiency.*

EXAMPLE I.—If 1,000 lbs. of water pass per minute through the drive pipe under a head of 6 feet, and 60 lbs. of water are delivered into the receiving tank, which is 87 feet above the check valve, what is the efficiency?

$$\text{Efficiency} = \frac{\text{work got out}}{\text{work put in}} \text{ (in the same time).}$$

$$\therefore \quad \text{Efficiency} = \frac{87 \times 60}{6 \times 1000} = \cdot 87$$

$$\text{Or,} \quad \textbf{Efficiency} = \textbf{87 per cent.}$$

If the length of the drive and rising pipes be considerable, and if there be many bends and much throttling of the passages, then the efficiency will thereby be reduced to a considerable extent. By a simple modification of the ram shown in the illustration, river or impure water may be made to raise spring or pure water; the two waters are separated by a diaphragm, and the pumping action actuates two valves, the one being a suction and the other a delivery valve.

Professor Irving P. Church in his text-book on *Hydraulic Motors* gives the following as the mode of action of the hydraulic ram:—

"The water in the drive-pipe having thus been brought to rest, but being still slightly compressed, an elastic recoil or rebound takes place and the pressure underneath the two valves C V and D V quickly falls to a low value, less than one atmosphere, so that the valve C V closes, while D V is opened, both on account of its weight and of *the pressure of the outer air*.

"RESULTS OF EXPERIMENTS ON A NO. 2 GOULD'S HYDRAULIC RAM, carried out by Professor Church at Cornell University in March, 1899.

No.	Whole Time in Minutes	Pul-sations per Minute.	Length of Stroke of Waste Valve in Closing.	Lbs. of Water Raised per Minute.	Height (H) through which Water is Raised, in Feet.	Lbs. of Water Passing through Waste Valve per Minute.	Fall or Working Head (h) of Waste Water, in Feet.	Efficiency of Ram. η.
1	12	80	$\frac{7}{8}''$	5·50	49·4	26·7	17	0·598
2 (*s*)	15	66	$\frac{7}{8}''$	2·30	56·4	21·8	10	0·595
3 (*s*)	16·7	98	$\frac{5}{8}''$	4·82	49·4	20·6	17	0·680

"In No. 1 the full weight of the waste valve D V, which was 6⅝ ounces, was operative, there being no provision to counterpoise a portion of it; and the length of its movement, or 'stroke,' was ⅞ inch. In the other two, Nos. 2 (*s*) and 3 (*s*), a light spring was employed, by the use of which the waste valve was virtually relieved of about one-half of its weight (though, of course, its 'mass' was practically unchanged). In No. 2 (*s*), the length of stroke of valve being the same as before, the efficiency is maintained at nearly 60 per cent., notwithstanding the fact that the ratio $H \div h$ is nearly double what it was in No. 1. This is doubtless due to the (relatively) quick closing of the valve. In No. 3 (*s*) the stroke has been made shorter with the effect of increasing the efficiency to 68 per cent., to which the decrease in the ratio $H \div h$ has also probably contributed somewhat. The pulsation is here very rapid, being 98 to the minute."

The Venturi Law.—This law states, "*that a fluid flowing through a conical pipe of gradually diminishing cross area loses head or pressure as it gains in velocity, and* vice versâ, *when flowing through a gradually expanding pipe or cone it loses speed as it gains in head or pressure.*" This fact is said to have been known to the ancient Roman hydraulic engineers It was, however, first established on a scientific basis by the Italian philosopher Venturi in 1796, when he was acting as Professor of Physics at the University of Bologna.

The Venturi Tube and Meter (*see the Frontis-plate*).—This tube, as now made for water mains, forms a part of the pipe line. It only differs therefrom in that, for a short distance, there is a cone at H C, coupled by a throat-piece T to the expanding cone V T. There is, therefore, no moving part in contact with the flowing water, and therefore no interruption to the supply. The exterior of the throat and of the inlet side are each provided with annular pressure chambers. These chambers at C_1 and C_2 communicate with the interior by means of small holes, which are bushed with vulcanite to prevent incrustation. The interior ends of these bushes are made flush with the inside of the pipe and throat, and, consequently, the pressure of water in the chambers at the cocks C_1 and C_2 is the same as that inside the throat T and the inside of the inlet end of main pipe respectively. Small copper pipes P_1 and P_2 convey these pressures up to the Kent recording apparatus. *

* This combination of the special throat-piece and surrounding pressure chambers, with the small holes from it into the throat, together with the same into the main inlet end of pipe, combined with piezometers or pressure gauges attached at C_1 and C_2, constituted the patent of Clemens Herschel, M.Inst.C.E., a distinguished American Hydraulic Engineer of Providence, R.I., U.S.A.

Theory of the Venturi Tube or the Principle of Action in the Double Cone.—The principle upon which this double cone acts is the same as that taken advantage of in the patent "Ferranti Steam Stop Valve."* And, therefore, both depend for their action upon the fundamental principle as explained by "Bernouilli's Theorem" at the beginning of Lecture IV. in this book—viz., when a perfect fluid is flowing in a *perfectly* smooth pipe (of uniform or of varying cross-section) it possesses *kinetic energy* in virtue of its *motion*, in addition to the *potential energy* due to its *position*, and *pressure energy* due to its *head* or *pressure.* Also, the *total energy* at any cross-section is the *sum of these three energies.*

Total Energy at each Cross-section.—Referring to Fig. 1

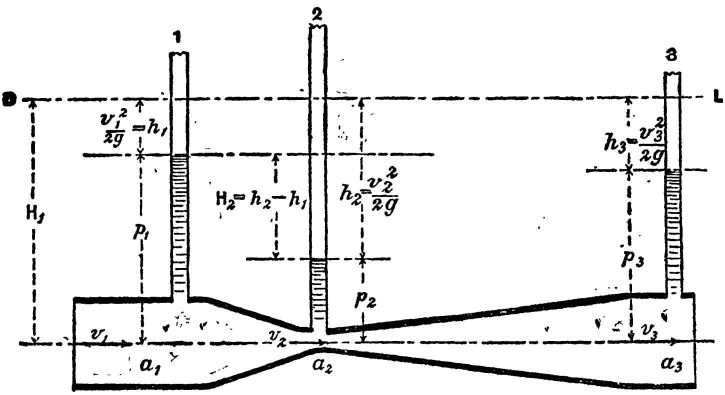


FIG. 1.—TO ILLUSTRATE THE THEORY OF THE VENTURI METER.

(which is drawn to illustrate the above principle), we see that the *velocity of the flow* of water as it just enters the Venturi tube or cone from the main pipe is marked v_1 (say, in feet per second). Also, that the height or *loss of head* is h_1, from the assumed datum line D L (which may *in reality* be the surface of the source of supply, such as a dam or reservoir). And, that the cross area of the main at the section 1 is a_1 square inches, whilst the *pressure* due to "*head*" at the cross-section is p_1, whilst the total *head* from the source of supply or datum line D L to the centre of the Venturi tube is marked H_1 on Fig. 1.

* For an illustrated description of "The Hopkinson-Ferranti Steam Stop Valve," see the 17th or later edition of Prof. Jamieson's *Text-Book on Steam and the Steam Engine, including Turbines and Boilers,* at Lecture XXVIII.

Now, remembering that the foregoing *principle* deals with the flow of a *perfect* fluid (such as water is assumed to be in the present case) through a *perfectly* smooth pipe, and knowing that the same weight of water W must pass each cross-section, not only of the main supply pipe, but also in the Venturi tube (if no leakage or eddies take place anywhere) there will be no resistance to the flow of the fluid. Consequently, the *total energy at section 1* will be—

Energy due to motion + energy due to position + $\left\{\begin{array}{c}\text{energy due to head}\\ \text{at section 1.}\end{array}\right.$

Or, $$\frac{W v_1^2}{2g} \quad + \quad W h_1 \quad + \quad W p_1.$$

In the same way, the *total energy at section 2* or *the throat* will be— $$\frac{W v_2^2}{2g} + W h_2 + W p_2.$$

Also, the total energy of the fluid as it leaves the Venturi tube will be— $$\frac{W v_3^2}{2g} + W h_3 + W p_3.$$

And, under the assumed *perfect* conditions, these *three* total energies must be equal to each other. Of course, we know that this is not the case, because we are neither dealing with a *perfect fluid* nor with a *perfectly smooth* pipe and tube, &c. But, the necessary corrections can be made for these imperfections (such as the *friction head* along the main supply pipe and in the Venturi tube or meter, &c.), if we get at first an expression to satisfy the fundamental principle.

Venturi Head, Throat Ratio, and Velocity.—Again, referring to the previous Fig. 1, we know from Vol. I. (of this series of Text-books) on *Applied Mechanics*, at Lecture XI., p. 273, that—

At Section 1—

The loss of potential energy = the kinetic energy evoked.

Or, $$W h_1 = \frac{W v_1^2}{2g}.$$

That is, $$h_1 = \frac{v_1^2}{2g} \text{ (as printed on Fig. 1).}$$

Also, at the Throat, Section 2—

$$W h_2 = \frac{W v_2^2}{2g}.$$

That is, $$h_2 = \frac{v_2^2}{2g} \text{ (as printed on Fig. 1).}$$

Now, neglecting all resistance to the flow of the water through the Venturi meter, the *loss of head* H_2, or drop h_1 to h_2, between sections 1 and 2 (or the "*Venturi head*" as it is called), is equal to the *loss of pressure* between those two points—viz., it falls from p_1 to p_2. But, it varies *inversely* as the velocities v_1 and v_2 at the cross-sections a_1 and a_2 in strict accordance with Bernouilli's Theorem.

$$\text{Hence,}\quad H_2 = (h_2 - h_1) = \left(\frac{v_2^2}{2g} - \frac{v_1^2}{2g}\right) = \frac{v_2^2 - v_1^2}{2g} = p_1 - p_2.$$

$$\text{Or,}\quad h_2 = H_2 + h_1 = \frac{v_2^2}{2g}\ \text{(see Fig. 1).}$$

$$\therefore\quad \textit{The throat velocity,}\ v_2 = \sqrt{2g(H_2 + h_1)}$$

$$\text{Hence,}\quad v_2 = \sqrt{2g\left(H_2 + \frac{v_1^2}{2g}\right)}.$$

Consequently, *the velocity at the throat varies as the square root of the "Venturi head"* H_2, *plus the loss of head* h_1 (*from source of supply*) *in producing the velocity of approach* v_1. But, these velocities *vary inversely* as the cross areas at a_1 and a_2.

$$\text{Or,}\ v_1 = \left(\frac{a_2}{a_1}\right)v_2: \qquad \therefore\quad v_1^2 = \left(\frac{a_2^2}{a_1^2}\right)v_2^2.$$

And (from above),

$$H_2 = \frac{v_2^2 - v_1^2}{2g} = \frac{\left(v_2^2 - \frac{a_2^2}{a_1^2}v_2^2\right)}{2g} = \frac{\left(1 - \frac{a_2^2}{a_1^2}\right)v_2^2}{2g} = \frac{\left(\frac{a_1^2 - a_2^2}{a_1^2}\right)v_2^2}{2g}.$$

$$\text{Hence,}\quad v_2^2 = 2g\,H_2\left(\frac{a_1^2}{a_1^2 - a_2^2}\right).$$

Or, *the throat velocity,*

$$v_2 = \left(\frac{a_1}{\sqrt{a_1^2 - a_2^2}}\right)\sqrt{2g\,H_2}.$$

That is, v_2 = constant for the Venturi meter $\times \sqrt{2g\,H_2}$.

But the ratio a_1/a_2 is termed the *throat ratio.*

Example.—What is the constant for the Venturi meter when the throat ratio is only 1/9?

$$\text{Here, the constant} = \frac{a_1}{\sqrt{a_1^2 - a_2^2}} = \frac{9}{\sqrt{9^2 - 1^2}} = \frac{9}{\sqrt{80}} = 1{\cdot}0062.$$

$$\text{Hence,}\quad v_2 = 1{\cdot}0062\sqrt{2g\,H_2}.*$$

* See *The Civil Engineers' Pocket-Book*, by John C. Trautwine, as published by John Wiley & Sons, New York, and Chapman & Hall, Ltd., London.

The size of the Venturi meter is reckoned by the size of the main pipe and not by the size of the throat.

Friction Head of the Venturi Tube.—With a velocity v_1 in the main pipe of 2 feet per second, and the throat ratio being 1/9, it will be found that the "Venturi head" is 5·16 feet and the *friction head* (or $h_3 - h_1$ in Fig. 1) is 0·79 of a foot. At a speed of 3 feet per second in the same main supply pipe (which is regarded as the maximum of good practice), the "Venturi head" will be 12·7 feet and the *friction head* 1·9 feet.

The Effect of Varying the Throat Ratio.—In Figs. 2 and 3 the

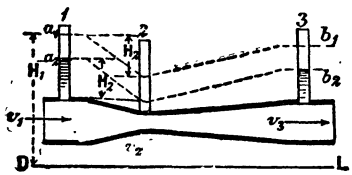


Fig. 2.

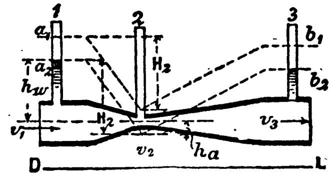


Fig. 3.

FIGS. 2, 3, AND 4 ILLUSTRATE THE EFFECT OF ALTERING THE THROAT RATIO.

dotted lines a_2 to b_2 join the vertical water surfaces of the centre points of the stand pipes 1, 2, and 3, whilst the dotted lines a_1 to b_1 indicate the pressure of the atmosphere, and, consequently, read absolute pressures from the datum lines D L, which may indicate sea level. In Fig. 2 a normal *throat ratio* has been taken, whereas, in Fig. 3, an abnormal one is depicted. In the latter case, all the water has disappeared from the stand pipe or piezometer 2 at the throat, and even a portion of water representing the atmospheric pressure has disappeared, thus leaving only a fraction to represent such pressure as remains in the throat.

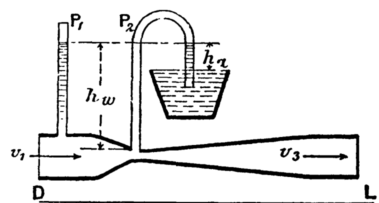


FIG. 4.—TO MEASURE THE VACUUM PRODUCED WHEN THE THROAT RATIO IS SMALL.

Fig. 4 shows a simple means of measuring the partial vacuum created by a high liquid velocity and an abnormally small throat. Here the height h_a of water or mercury which rises in the right-hand leg of the bent tube indicates, as in an ordinary barometer, the extent of the vacuum produced at the throat. Hence $(h_w + h_a)$ = total loss of head or "Venturi head" between the piezometers P_1 and P_2.

The Application of the Venturi Law to the Measurement of Water.—This has been successfully carried into effect by the Clemens Herschel (mentioned in the previous footnote), the eminent American hydraulic engineer. An exhaustive series of experiments was made by him, which has established the absolute reliability of the principle under every conceivable condition.* Tubes varying from $\frac{1}{4}$ inch to 9 feet in diameter, have all answered exactly to the Venturi law. Numerous careful tests against meter, weir, and tank measurement have invariably confirmed the practical truth of this law. In a seventeen months' test, carried out at New Jersey, U.S.A., with thirteen Venturi meters and with an average daily consumption of 40,000,000 gallons, the discrepancy between the total registration of the two receiving meters and that of the eleven meters was only 0·5 per cent.

When the immense volume of water measured is taken into account, the accuracy of the result is extraordinary. This is certainly the largest and longest test that has yet been made with the Venturi meter, but it is only another instance of that which has been proved both experimentally and practically with the Venturi meters of all sizes, from 3 inches to 9 feet—namely, that the law is absolutely reliable.

The Uses of the Venturi Meter.—"In its larger sizes the meter is practically without a competitor. It accomplishes rapidly what has previously been obtained rarely, or only approximately. It enables large volumes of water to be measured easily and accurately. It determines the leakage of pumps, mains, or reservoirs, and offers to waterworks' engineers a means of determining accurately, and at little cost, the total quantity of water drawn, the quantity used by large mills, adjacent towns, or other large consumers. It can be placed in a flume or channel, and is well adapted for use in irrigation and in sewerage work."

This instrument measures water flowing under pressure in mains of any capacity, without offering any obstruction in, or causing any deviation to, the direct pipe line; and this is accomplished without the employment of any moving part whatsoever in contact with the flowing water. It can also be applied, with the addition of Kent's Recorder, to measure irrigation waters flowing either in open channels or in pipes. In mining operations it has been used to measure slimes and cyanides, and, generally, hot water or feed-water for steam

* See "The Venturi Meter," a paper which was read before the American Society of Civil Engineers by Mr. Herschel, and which gained for him the Rowland Prize.

boilers. It acts as a capital inspector of pumping stations, detecting any want of efficiency in the plant and indicating the "slip factor," thereby drawing the attention of the engineer in charge to any defects and need for repairs.

Kent's Recorder or Register* (*see the Frontis-plate*).—If the difference of pressure or "Venturi head" between the throat T at the cock C_1 and the up-stream cock C_2 has a constant relation to the discharge from the tube, *it is only necessary to multiply the square root of this pressure* by the ascertained coefficient and by time in order to obtain the total quantity of fluid passed.

The register itself consists of two parts. *First*, of a mercurial U-tube in the shape of the mercury chambers M_1, M_2, which, being connected with the throat and up-stream by the pipes P_1 and P_2, brings in the element of "Venturi head." And, *second*, of the clockwork and gear controlled thereby, which supplies the element of time. The connection between the pressure and time is established by means of floats F_1, F_2 resting on the mercury in the U-tube at F_1 and F_2. In the combined instrument, as illustrated in the Frontis-plate, we have a diagram D giving the rate of flow, as well as a pressure recorder P R, and counter V C showing the total quantity passed. The floats F_1, F_2, inside M_1, M_2, are made of iron. They have a circular collar or bead projecting from their sides, so that the collars just sit on the surface of the mercury. In this way more than in any other the floats are rendered susceptible to a rise or fall of pressure. These floats F_1, F_2 carry light racks, gearing into pinions, which convey the motion to the corresponding racks R_1, R_2 placed outside the tubes. The latter have light rods (shown in the Fig.), which pass up to the clockwork mechanism. The one on the left belongs to the diagram recorder D, and supports from its upper end a pen guide G, and carriage. The rod on the right extended from R_2 regulates the amount of registration by the Venturi meter counter V C. It is fitted with a small carriage, resting on the surface of the inner drum, which rotates at a uniform speed. This drum is recessed according to the square root principle. It

* This Recorder was patented by Walter G. Kent, Managing Director of George Kent, Limited, of 199 High Holborn, London, W.C. They have kindly supplied the Author, specially for this book, with the new Frontis-plate as actually arranged and connected-up in practice to the Venturi Meter Tube. The *Index to Parts* has been printed with index letters in accordance with the particular method devised by the Author and adopted in each of his ten text-books. The student should go carefully over this index and see that he finds out each part of the Recorder *before* reading the following description.

throws the counter in and out of gear for a portion of its revolution, according to the rate of flow. At no flow, the roller is on the raised portion at the top of the drum, while, at maximum flow, it rests on the recessed portion at the bottom of the drum for the whole of the revolution. Intermediate points are in proportion to the square root of the "Venturi head." That portion of the apparatus belonging to the diagram is so similar to other recording instruments that it requires no special description. It may be fitted either with a daily or with weekly diagrams. It is ruled with horizontal lines showing the rate of flow of the liquid, and with vertical lines giving the hours and days as determined by the clock O and its pendulum P. It will be seen that the height of the pen carriage is directly proportional to the height of mercury in the left leg, M_1, of the U-tube—in other words, to the "Venturi head" or difference of pressure in the throat T at C_1 and the main inlet pipe at the cock C_2. Thus the registration is direct and simple.

In order to comply with *the hydraulic law* which makes the quantity of liquid discharged through a pipe *vary directly as the square root of the head*, it is necessary to *square-root* the "Venturi head" (which is given by the right leg of the mercurial tube) *before multiplying it by time to obtain the total quantity passed.* That is done automatically by the recorder and its integrating scale.

Pressure Recorder Attachment.—It will be seen from the Frontis-plate that the pressure recorder or P R scale is fixed to the top of the Venturi diagram drum D, while the pen movement is carried on the side pillar.

This addition is only applicable to instruments of the mercurial type. By its use, a diagrammatic chart of the pressure in the main at the point where the Venturi tube is fixed is obtained. As this pressure diagram is synchronous with that of the rate of flow given (diagram D) by the meter, a record is provided, not only of the pressure at all hours, but also of the friction or loss of head in the main resulting from certain known velocities. Therefore, an indication is obtained of the state of the main. If, for instance, with a certain increase in velocity, there was a loss of pressure in excess of that due to the friction of a fairly clean main of the same diameter, it would be obvious that the main in question was becoming choked or dirty.

Friction Curve.—The following diagram of a friction curve is both interesting and instructive. After what has been said in the text, this diagram will be self-explanatory, since all the necessary data is given on the figure.

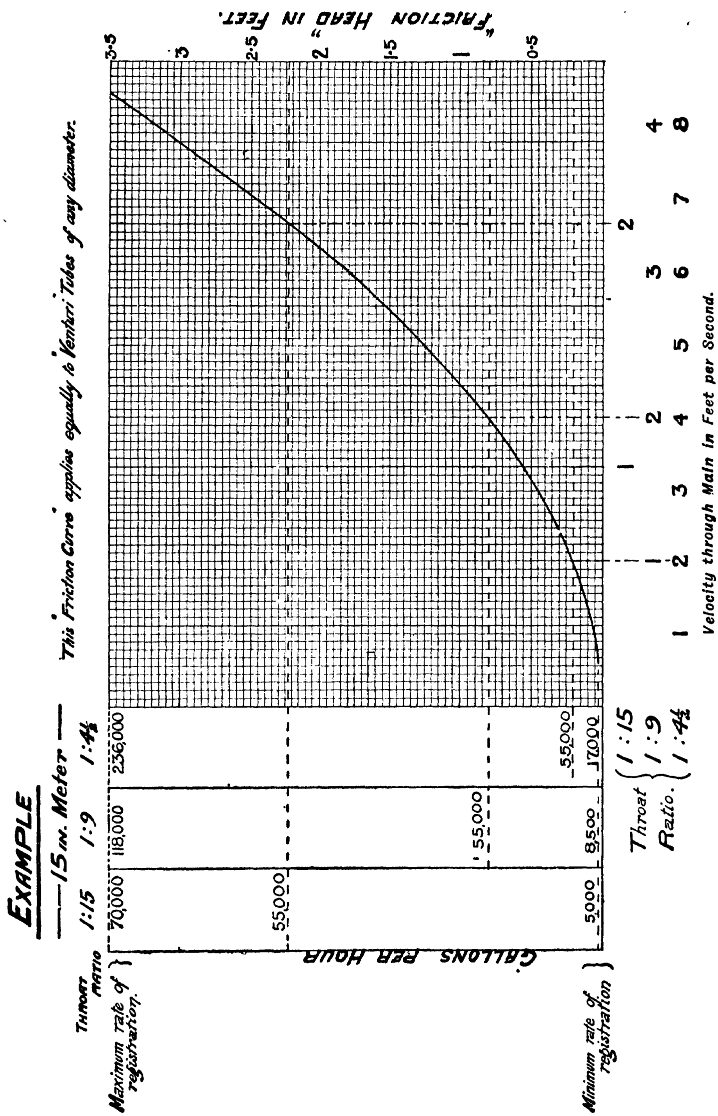


DIAGRAM SHOWING LOSS OF HEAD DUE TO VELOCITIES THROUGH VENTURI METER TUBES OF THE

*The Author is indebted for the loan of the Plate from which the above Fig. is printed to George Kent & Co., Ltd., High Holborn, London, W.C.

Lecture III.—Questions.

1. Prove the law for changing p, v, and h along a stream line in a frictionless fluid. Apply the law to find the funnel shape of the surface of water in a basin from which the water is flowing by a central hole.

2. Prove the law for changing p, v, and h along the stream lines in a frictionless fluid. Apply, neglecting change of level, to the case of adiabatic flow of air from one vessel to another through a small orifice, and deduce the rule for maximum quantity flowing.

3. Describe the action of a jet pump, or of a good form of injector.

4. Describe the construction and action of a Venturi meter, and deduce a formula for the discharge from a meter of this type. A Venturi meter in a 30-inch pipe line has a throat diameter of 12 inches, and the mean difference of head between the large and small sections was found to be 14·6 feet of water during a period of 24 hours. Calculate the average discharge from the meter per hour.

5. A pipe tapers to one-tenth its original area, and then widens out again to its former size. Calculate the reduction of pressure, at the neck, of the water flowing through it, in terms of the area of the pipe and the velocity of the water. Why is this reduction of pressure a gauge of the discharge?

6. Describe the hydraulic ram, and give a formula for its delivery at a height h feet above it, the supply being Q cubic feet per second coming from a height of H feet.

7. A slanting pipe, 2 inches in diameter, gradually enlarges to 4 inches. The pressure at a given section in the 2-inch pipe is 25 lbs. per square inch, and the velocity is 8 feet per second. Calculate the pressure in the 4-inch portion 14 feet below. *Ans.* 31·2 lbs. per square inch.

8. A horizontal tube is tapered slowly from a diameter of 15 inches to a diameter of 6 inches. Neglecting friction, find the difference in pressure at the two sections when the discharge is 60,000 gallons per hour. *Ans.* 1·214 lbs. per square inch.

LECTURE III.—I.C.E. QUESTIONS.

1. A conical pipe varying in diameter from 4 feet 6 inches at the large end to 2 feet at the small end forms part of a horizontal water main. The pressure-head at the large end is found to be 100 feet, and at the small end 96·5 feet. Find the discharge through the pipe. *Ans.* Velocity = 15·3 feet per second; discharge = 48·2 cubic feet per second.

2. Give the expression for the total energy of a pound of water flowing in a pipe; if the velocity of the water is 5 feet per second, its pressure 60 lbs. per square inch, and the height of the axis of the pipe at the point in question is 10 feet above the datum, find its value.

3. A pipe 9 inches diameter gradually contracts to 4 inches diameter, its axis being level. The tops of the two legs of a U-tube containing mercury are connected by tubes to the centres of the two sections, and it is found that the difference in the level of the mercury in the two legs is 6 inches. What quantity of water is flowing through the pipe? The specific gravity of mercury is 13·6.

4. Explain how the principle of the conservation of energy is applied to obtain in expression for the energy of water due to its elevation, pressure and velocity. A water-main discharging 300 gallons per minute has a diameter of 5 inches at a gauge point where the pressure is 45 lbs. per square inch. At a second gauge point 25 feet below the first, and with a reduced diameter of 4 inches, the pressure in the pipe is 40 lbs. per square inch. Calculate the loss of head between the two sections due to friction and other resistances.

5. What is the connection between the velocity and pressure at different parts of a smooth horizontal pipe of varying diameter and which is running full of water, all resistance being neglected? If the pressure where the diameter is 2 inches is 500 lbs. per square inch and velocity 4 feet per second, what will be the pressure where the diameter is 1 inch? *Ans.* 498 lbs. per square inch.

6. State and prove "Bernouilli's Theorem" relating to the steady motion of incompressible fluids, and apply it to find the discharge through a Venturi water-meter.

7. Sketch a practical form of hydraulic ram for raising automatically to a considerable height a portion of any volume of water derived from a lesser height, and explain the points upon which satisfactory working depends.

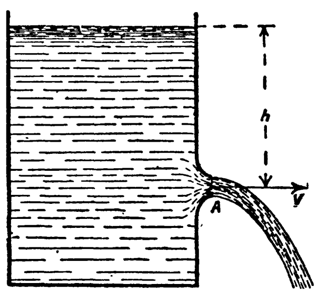
LECTURE IV.

FLOW OF WATER, &c.

CONTENTS.—Velocity of Efflux and Flow of Water from a Tank—Rectangular Gauge Notch—Thomson's Triangular Notch—Measurement of Head—Measurement of Large Streams—Horse-Power of a Stream—Pitot Tube—Free and Forced Vortex—Centrifugal Pressure—Reaction of a Jet—Re-entrant Orifice—Flow in Pipes—Hydraulic Mean Depth—Questions—Solutions to I.C.E. Questions.

WE have considered the general principles governing the flow of water; we come now to consider the flow of water over weirs and through orifices, the resistance to flow in pipes and channels, and kindred problems.

Velocity of Efflux and Flow of Water from a Tank.—Consider a jet of water issuing from a small circular orifice A at a



EFFLUX OF WATER FROM A CIRCULAR ORIFICE.

depth h below the free surface of the water in the tank. If we take the datum line at the orifice, then inside the vessel, where the water is still, the energy is entirely potential and equal to $m g h$; whereas, it is all kinetic just outside the opening and amounts to $\frac{1}{2} m v^2$.

Hence, $\frac{1}{2} m v^2 = m g h$; or, $v^2 = 2 g h$;

i.e., $$v = \sqrt{2 g h}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (\text{VI})$$

It will be observed from this equation (VI) that the velocity

v is the same as that attained by a body in falling freely from a height h; and further, that it would be the same even if the water had no free surface, so long as the pressure at the level of the orifice was equal to that due to a head h.

If there be a loss of head f due to friction and eddies formed by the water in passing through the orifice,

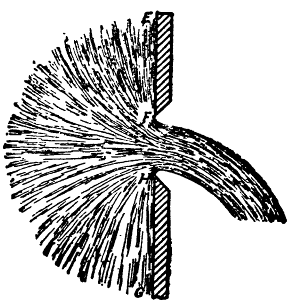
Then, $\frac{1}{2} m v^2 = m g (h - f)$; or, $v = \sqrt{2g(h-f)}$. (VII)

If, a = The cross area of the orifice in square feet.
r = „ radius of the orifice in feet.
v = „ average velocity of the water in feet per second.
h = „ head of the water in feet.
g = „ acceleration due to gravity or 32·2 feet per second per second.
And Q = „ quantity of water flowing out from the orifice in cubic feet per second.

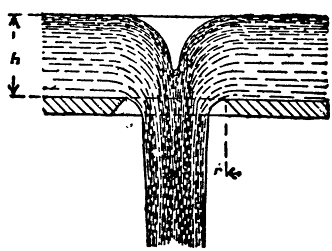
We find from equations (V) and (VI) that :—

$$Q = a v = a \sqrt{2 g h} \text{ cubic feet per second.}$$

This formula is very nearly true for a small tapered opening, but with a flat one, as shown in the next figure, the cross area of the



Efflux of Water from a Vertical Flat Orifice.



Efflux of Water from a Horizontal Flat Orifice.*

stream where the stream lines are parallel, at a short distance outside the opening, is less than that of the orifice. The ratio of these two areas is called the *coefficient of contraction*, and it is found experimentally for a flat opening to be 0·64. The contraction is caused by the water flowing along the inner flat surfaces E F and G H and then leaving them at a tangent. It has also been found by experiment that for a sharp-edged orifice the velocity v is only

* By mistake the figure has been drawn with a vortex, but when measuring water we must prevent the formation of a vortex by putting in radial blades.

$0.97 \sqrt{2gh}$. Hence, the actual flow for *a small circular orifice* will be :—

$$Q = 0.64\, a \times 0.97 \sqrt{2gh} = 0.62\, a \sqrt{2gh}$$

$$\therefore \quad Q = 0.62\, \pi r^2 \sqrt{2gh} \text{ cubic feet per second.} \qquad \text{(VIII)}$$

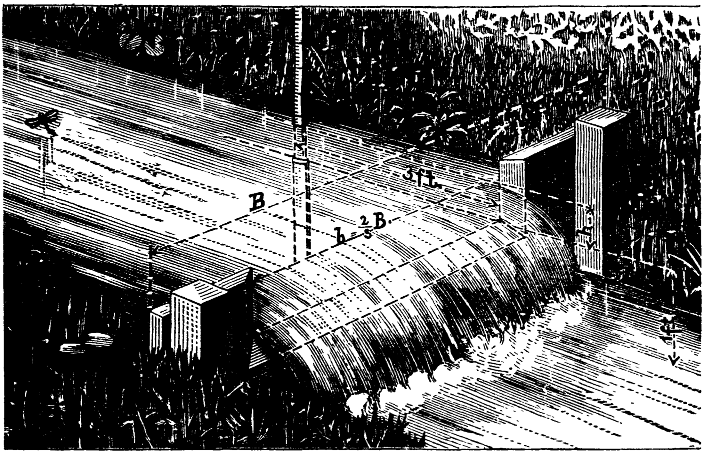
In measuring a small flow of water by this method, it is run into a tank having a carefully made clean cut orifice of known area, until the surface level is just sufficiently above the outflow to cause the water to run out as fast as it runs into the tank. This difference of level, or head h, is measured and the above formula applied.

Accurate results cannot be obtained from a large circular opening placed in the side of a tank, because the parts of it would have different depths from the free surface, and consequently the water would have different velocities at these parts. If we, however, put the orifice in the bottom of the tank, as shown in the right-hand figure, then the velocity will be approximately the same at all parts of the opening, and we can enlarge it so as to measure a much greater flow of water.

Measurement of a Flowing Stream.—In order to ascertain the available power from a stream or river, as well as to test the efficiency of a hydraulic installation, it is of the first importance to determine the rate of flow—*i.e.*, the number of cubic feet or gallons of water passing a given point per unit of time. At first sight, this would appear to be a very simple matter; but, as will be shown, it is not so easy to do so with accuracy, for several special precautions have to be observed and constants obtained.

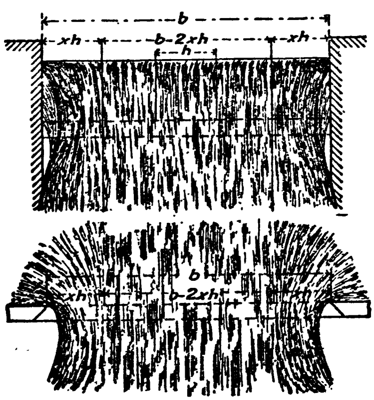
Rectangular Gauge Notch.—When we have to measure large quantities of moving water, then such orifices as we have previously dealt with are quite unsuitable. In such cases, it is usual to pass the whole of the water over a special form of weir or gauge notch. This consists of a board placed across the stream between stakes and carefully puddled, so that all the water must flow over it. The top bevelled edge of this board must be well above the surface of the water on the delivery side. The length of the notch is usually less than the width of the stream, and the board should be continued from each end in a straight line (as shown in the perspective view) so as to give definite conditions; or parallel guides may be placed in the stream before the gauge board. In the former case, the inflow of water at the ends of the notch board causes end contraction, while in the latter, this effect is avoided. In each case, contraction of the stream is caused by the water flowing upwards from E to F, as shown by the sectional figure a few pages further on. Further, the surface directly over the notch board is

lower than that of the still water in the pond above the board. If, however, we neglect these effects we see that with different lengths



PERSPECTIVE VIEW OF A RECTANGULAR GAUGE NOTCH.

of notch board the total flow will be proportional to its length. Now, consider any horizontal strip of the cross section of the stream over the notch (say, one-thousandth of its total depth); then, if we increase the depth of the stream over the notch, the vertical width of this strip and consequently its area will be proportionally increased, as well as its depth below the free surface of the water. But the velocity of the water passing through this strip varies as the square root of its depth, and the quantity as the area multiplied by the velocity. This result will hold good for each elementary strip, and will, therefore, apply to the whole stream.



FRONT VIEW AND PLAN OF A RECTANGULAR GAUGE NOTCH.

Hence, $Q \propto a\,v \propto h \times \sqrt{h} \propto h^{\frac{3}{2}}$ for different depths;

And, $Q \propto b$ for different breadths of stream at the notch;

$\therefore$ $Q = k\,b\,h^{\frac{3}{2}}$ for a rectangular notch. . . . . . (IX)

Here, k is a *coefficient of discharge* which must be found by experiment. This equation, however, would not give us accurate results if the proportions of the stream passing the notch were much different from that used to determine the constant k. With a very long shallow notch a considerable error will arise from the fact, that the water may adhere to the horizontal bevelled edge, and with a very deep narrow notch a similar effect would be produced by the bevelled sides. The above formula is often used and is quite correct for similar streams; but if the flow is variable, the depth will change with the flow of the water while the breadth remains constant, so that the proportions of the stream obtained with different flows in a gauge notch of this kind are not the same.

The late Professor James Thomson showed how we may obtain a formula which will apply to all ordinary proportions of rectangular notches. In the central part of the stream the lines of flow are practically parallel and unaffected by the sides of the notch. Consequently, the water passing through this part will be proportional to its breadth. Suppose the influence of each end to extend perceptibly to a distance $x\,h$ from it. Then the breadth of the central part will be $b - 2\,x\,h$. Consider a portion of this central part whose breadth is equal to h. This will be a square and therefore similar for different sizes of stream. Hence, since the area is proportional to h^2 and the velocity to $\sqrt{h}$, the flow through this square will be :—

$$k_1 \times h^2 \sqrt{h} \text{ or } k_1\,h^{\frac{5}{2}}$$

where k_1 is a constant.

Consequently, the flow through the whole of the central part $(b - 2\,x\,h)$ will be :—

$$\frac{b - 2\,x\,h}{h}\,k_1\,h^{\frac{5}{2}} = k_1\,(b - 2\,x\,h)\,h^{\frac{3}{2}}.$$

If we now imagine the two side portions to be placed together, we will get another stream which will be of similar form whatever its actual size may be; for, it will always be a rectangle of depth h and length $2\,x\,h$. Consequently the flow through this will be:—

$$k_2 \times 2\,x\,h \times h^{\frac{3}{2}} = 2\,k_2\,x\,h^{\frac{5}{2}}$$

where k_2 is another constant.

Hence, the flow of the whole stream will be :—

$$Q = k_1 (b - 2\,x\,h)\,h^{\frac{3}{2}} + 2\,k_2\,x\,h^{\frac{3}{2}}$$

$$\therefore \qquad Q = k_1 \left\{ b - \frac{2\,x\,(k_1 - k_2)}{k_1}\,h \right\} h^{\frac{3}{2}}.$$

If we write c for $\frac{x\,(k_1 - k_2)}{k_1}$ which is a constant, we get:—

$$Q = k_1 \{ b - 2\,c\,h \} h^{\frac{3}{2}}. \quad . \quad . \quad . \quad . \quad . \quad (X)$$

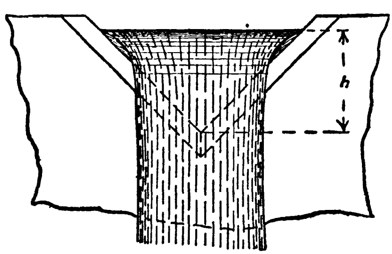
If one of the sides of the stream had a guide board we should have had x throughout instead of $2\,x$, and therefore $c\,h$ in our final results instead of $2\,c\,h$. If both sides of the stream be guided, this term would disappear and we would get the former result.

Mr. Francis, an American engineer, deduced the following empirical formula from a large number of experiments which he made :—

$$Q = 3{\cdot}33 \left(b - \frac{1}{10} n\,h \right) h^{\frac{3}{2}} \text{ cubic feet per second.} \qquad (X_a)$$

Here n is the number of end contractions (viz., 2, 1, or 0, as explained above), and the units employed are feet and seconds. It should be noted that this equation is of the same form as the previous one, and that neither is applicable to a notch whose length is less than $2\,x\,h$.

Thomson's Triangular Notch.—Professor James Thomson proposed and used a gauge notch in the form of a right-angled isosceles triangle with its sides equally inclined to the vertical. It has the advantage of giving a similar form of stream whatever may be the size of the notch or the height of the water passing through it, and is, therefore, more accurate for measuring variable streams. As, however, less water is passed for a given height than with the rectangular notch, it is not so convenient for large flows; but by cutting a number of such notches, side by side like the teeth of a saw, considerable quantities of water may be dealt with.



THOMSON'S TRIANGULAR GAUGE NOTCH.

If we consider corresponding elements of two such notches, we see that their areas are proportional to the square of their depths,

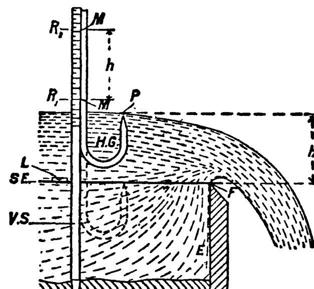
whilst, as before, the velocities of the water are as the square roots of the depths. Hence, the flow through a **V** notch will be :—

$$Q = k \times h^2 \times \sqrt{h} = k\,h^{\frac{5}{2}} \text{ cubic feet per second.} \qquad \text{(XI)}$$

In this case, the *coefficient of discharge* k, has been found by careful experiment to be 2·64.

Measurement of Head.—When using either of the previously mentioned notches for determining the flow of a stream or river we must ascertain the head h, with great accuracy. This may be done by aid of a level, straight edge, graduated staff, and a bent wire or hook-gauge in the following manner :—

Drive the vertical stake V S, into the bed of the stream at a position above the notch where the surface has no appreciable velocity. To obtain such a position the pond above the weir should not be too small. Level a straight edge S E, by the level L, with its lower edge resting on the inner edge F, of the bevelled board and on the point of the hook-gauge. Note the position R_1 on V S, opposite a mark M, on the longer limb H G. This gives us once for all the zero from which to reckon h.



APPARATUS FOR MEASURING HEAD OF WATER.

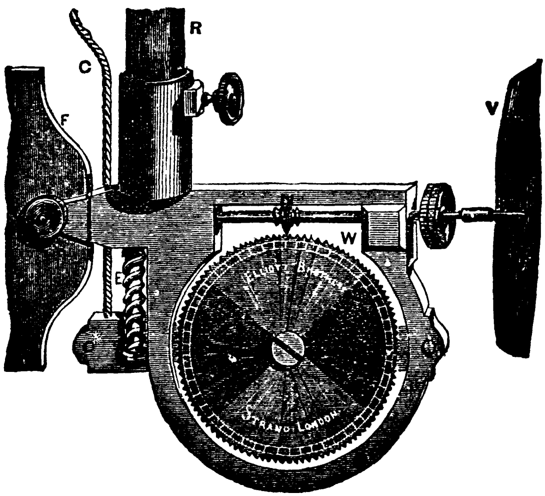
When the water is flowing in the normal condition for making the test, raise H G until the point P just touches the surface of the water and note the reading R_2 on V S opposite M, as shown by the full outlined hook in the figure. The difference between this reading and the former one gives the head h.

As may be seen from the previous formulæ any mistake made in determining h will produce a larger percentage error in the result with the **V** and rectangular notches than with an orifice in the bottom of a tank. The latter is, therefore, preferred where great accuracy is desired and the quantity of water is not too large, such as when measuring the circulating water used by a steam engine.

Measurement of Large Streams.—When it is inconvenient or impracticable to place a weir gauge in a river, then the flow may be estimated by measuring the cross-section, a, of the river and finding its mean velocity, v, at that section :—

Then, as before, $\qquad Q = a\,v.$

To do this, a number of equidistant points are marked along a rope which is then stretched across the river at right angles to the direction of the current. By means of a graduated pole the depth at each of these points is ascertained from a boat. The results are then plotted to scale and give us a cross section of the river and an estimate of its sectional area. The surface velocity at midstream may be roughly found by noting the time taken for a float to move a given distance down stream, and the *mean* velocity may be taken as 0·65 of this. It is, however, much more accurate to ascertain the velocity at a number of points of the section by means of a current meter and then calculate the mean value. The following illustration shows a current meter



ELLIOTT'S CURRENT METER.

made for this purpose by Messrs. Elliott Brothers, London. It has a screw-shaped vane V, which is rotated by the water flowing past it. The revolutions of this vane are counted by the wheel W, which is driven by a worm on the same spindle as the vane. When the apparatus is immersed by means of the rod R to the required depth, with the vane pointing up stream, the cord C is pulled up and kept tight for a definite interval of time. This cord is attached to the end of a lever which carries the bearings of the counting wheel and is pushed down by a spring E. The wheel only gears with the worm when the cord is pulled, and the reading gives the number of revolutions of the vane from which the velo-

city of the water may be deduced. F is a rudder to keep the apparatus pointing directly upstream.

***Measurement of Water.**—Since it is essential to ascertain as accurately as possible the quantity of water available for driving a turbine, we furnish descriptions of the various methods most commonly used for measuring the flow of a stream. One or other of these will be found applicable to almost any stream, and, if circumstances permit, it is advisable to try more than one, so as to check the results obtained by each method.

I. *By the Velocity of the Water and Sections of the Stream.*—Choose a length of the stream, say about 50 or 100 feet, along which the section is as uniform as possible, and find the area of the section by multiplying the width by the average depth. By taking several sections in the chosen length and determining the average section, a nearer approach to accuracy is obtainable. A stake should be fixed at each end of the measured length, and a float must be thrown into the middle of the stream, a little above the first stake, and the time noted which it takes to pass from one stake to the other. This should be repeated several times, and the average time taken, so as to get a more accurate result. The best float is a bottle partly filled with water, so that it may float upright.

The theoretical quantity of water flowing per minute is found by multiplying the area of the stream by the measured length and also by 60, and dividing the product by the time (in seconds) taken by the float. From the result obtained a deduction, amounting to about 15 to 20 per cent. for earthen banks, must be made to allow for loss of velocity through friction on the sides and bottom, in order to arrive at the actual rate of flow.

Example.—The sectional area of a stream is 20 square feet, and a float passes over a measured length of 90 feet in 36 seconds; hence $20 \times 90 \times 60 \div 36 = 3{,}000$ cubic feet of water per minute theoretically. Or, an actual quantity of about 2,400 cubic feet per minute after allowing 20 per cent. for friction.

II. *By a Notched Board.*—The amount of flow can now be found by referring to the following table, which gives the discharge in cubic feet per minute per inch length of notch for depths from 1 to $18\frac{7}{8}$ inches. This table gives very accurate results in most cases, but is not reliable for very small quantities.

* I am indebted to W. Günther & Sons, of Oldham, the well-known Water Turbine Manufacturers and Experts, for these data and tables.

TABLE OF DISCHARGE OVER RECTANGULAR NOTCHED BOARDS PER INCH LENGTH OF WEIR.

Depth (h) on Weir in Inches.	0	$\frac{1}{8}''$	$\frac{1}{4}''$	$\frac{3}{8}''$	$\frac{1}{2}''$	$\frac{5}{8}''$	$\frac{3}{4}''$	$\frac{7}{8}''$	Depth (h) of Weir in Inches.
1	·40	·47	·55	·65	·74	·83	·93	1·03	**1**
2	1·14	1·24	1·36	1·47	1·59	1·71	1·83	1·96	**2**
3	2·09	2·23	2·36	2·50	2·63	2·78	2·92	3·07	**3**
4	3·22	3·37	3·52	3·68	3·83	3·99	4·16	4·32	**4**
5	4·50	4·67	4·84	5·01	5·18	5·36	5·54	5·72	**5**
6	5·90	6·09	6·28	6·47	6·65	6·85	7·05	7·25	**6**
7	7·44	7·64	7·84	8·05	8·25	8·45	8·66	8·86	**7**
8	9·10	9·31	9·52	9·74	9·96	10·18	10·40	10·62	**8**
9	10·86	11·08	11·31	11·54	11·77	12·00	12·23	12·47	**9**
10	12·71	13·05	13·19	13·43	13·67	13·93	14·16	14·42	**10**
11	14·67	14·92	15·18	15·43	15·67	15·96	16·20	16·46	**11**
12	16·73	16·99	17·26	17·52	17·78	18·05	18·32	18·58	**12**
13	18·87	19·14	19·42	19·69	19·97	20·24	20·52	20·80	**13**
14	21·09	21·37	21·65	21·94	22·22	22·51	22·79	23·08	**14**
15	23·38	23·67	23·97	24·26	24·56	24·86	25·16	25·46	**15**
16	25·76	26·06	26·36	26·66	26·97	27·27	27·58	27·89	**16**
17	28·20	28·51	28·82	29·14	29·45	29·76	30·08	30·39	**17**
18	30·70	31·02	31·34	31·66	31·98	32·31	32·63	32·96	**18**

Example.—The depth of water (h) flowing over a notched weir 54 inches wide is 15½ inches; what quantity of water is passing down? Referring to the table, look for 15 inches in the first or last columns and follow the table across until the column headed ½″ is reached; the number there (24·56) is the rate of discharge in cubic feet per minute *per inch of weir length,* therefore the full quantity of water discharged per minute is $24{\cdot}56 \times 54 = 1{,}326$ cubic feet per minute.

III. *By Discharge through a Sluice or Shuttle.*—This method is very convenient in places where there is a shuttle already fixed in the bye-pass, and giving an unobstructed passage to the discharge of the water. The shuttle is raised until it exactly passes all the water coming down the stream. The width and height of the opening must then be measured, together with the exact depth of water from the surface to the centre of the opening. The following table gives the actual quantity of water passing per square inch of shuttle opening, under heads varying from 1 to 60 inches:—

TABLE OF DISCHARGE PER SQUARE INCH OF AREA THROUGH SLUICES OR ORIFICES.

Head in inches,	1	2	3	4	5	6	7	8	9	10
Cub. ft. per min.,	0·62	0·87	1·16	1·22	1·37	1·50	1·62	1·73	1·84	1·94
Head in inches,	11	12	13	14	15	16	17	18	19	20
Cub. ft. per min.,	2·03	2·12	2·21	2·29	2·37	2·45	2·53	2·60	2·67	2·75
Head in inches,	21	22	23	24	25	26	27	28	29	30
Cub. ft. per min.,	2·82	2·88	2·93	3·00	3·06	3·12	3·18	3·24	3·30	3·35
Head in inches,	31	32	33	34	35	36	37	38	39	40
Cub. ft. per min.,	3·41	3·46	3·52	3·57	3·63	3·68	3·72	3·77	3·82	3·88
Head in inches,	41	42	43	44	45	46	47	48	49	50
Cub. ft. per min.,	3·93	3·97	4·02	4·06	4·11	4·16	4·22	4·24	4·28	4·33
Head in inches,	51	52	53	54	55	56	57	58	59	60
Cub. ft. per min.,	4·37	4·41	4·45	4·50	4·54	4·58	4·62	4·66	4·70	4·74

Example.—A shuttle 40 inches wide and raised 18 inches exactly passes the stream with a depth of 28 inches above the centre of the opening. Referring to the table we find that under this head the shuttle passes 3·24 cubic feet per minute per square inch of area. And, as the area is $40 \times 18 = 720$ square inches, the actual quantity of water available is $3{\cdot}24 \times 720 = 2332{\cdot}8$ cubic feet per minute.

Horse-Power of a Stream.—After having obtained the quantity of water flowing in a stream, we have only to measure the available head in order to find its horse-power H.P. The head may be measured in feet by aid of a surveyor's level and staff. Then, if w be the weight of a cubic foot of water, and W the weight of the total flow of water per second, we get :—

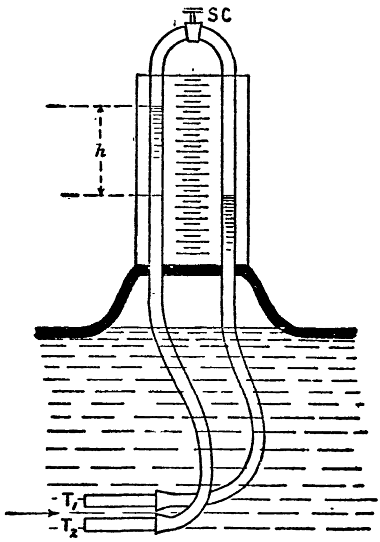
$$Q = a\,v. \quad \text{And } W = Q\,w.$$

$$\textbf{Hence,} \quad \text{H.P.} = \frac{\text{ft.-lbs. per second}}{550} = \frac{h \times Q\,w}{550}.$$

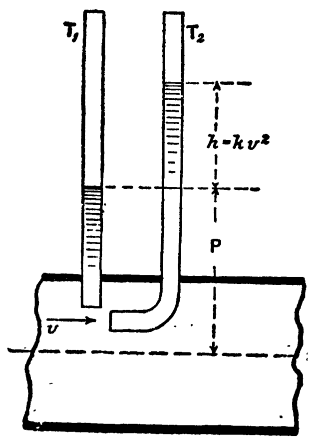
$$\therefore \quad \text{H.P.} = \frac{62{\cdot}5}{550} \times h\,Q = 0{\cdot}114\,h\,Q = 0{\cdot}114\,h\,a\,v. \qquad \text{(XII)}$$

The result found by Eq. (XII) is the theoretical or total horse-power available. But as no turbine or other water motor can utilise the whole of this power, a deduction varying from 20 to 25 per cent. must be made to find the actual or effective horse-power which the turbine is capable of developing.

Pitot Tube.*—Another apparatus which can be used for determining the velocity at a point in a flowing stream of a liquid or a gas, even when the stream is of small dimensions, such as that passing along a small pipe, is called a "Pitot tube." In its simplest form, as originally proposed by Mr. Pitot in 1732, it consists of a glass tube open at both ends and bent in the shape of the letter L. One leg of the L was held horizontal under water, with its open end facing the current; and the



LINE DIAGRAM OF PITOT'S TUBE, WHICH INDICATES THE VELOCITY OF WATER.



LINE DIAGRAM TO ILLUSTRATE THE FERRIS-PITOT METER.

velocity, v, at the point where it was placed was measured by the vertical height to which the water rose in the vertical leg above the surface of the stream.

The accompanying line diagram shows the Pitot tube, as developed by M. Darcy and Prof. S. W. Robinson, and consists of *two* horizontal glass or metal tubes, T_1 and T_2, of very small bore, placed side by side in the current and pointed up-stream.

* I am indebted to the 19th edition of *The Civil Engineer's Pocket-Book*, by John C. Trautwine, for this description of the Pitot Tube and the Ferris-Pitot Meter.

Tube T_2 receives the current in its open up-stream end, while T_1 is closed at its up-stream end, and has small *lateral* openings only. The other end of each tube communicates, by means of small metal or rubber piping, with one leg of an inverted U-shaped glass gauge fixed in a boat or on shore. For convenience, the two flexible pipes may be joined together into one double pipe.

By sucking through the top stop-cock, S C, water is drawn up to any convenient height in the two legs of the gauge. And when placed in a current of water, the difference, h, in the heights of the two columns is such that, $v = \sqrt{2gh}$.

No corrective coefficient is required with this simple and accurate instrument. It will even measure velocities as low as 4 inches per second.

In practice, the tubes, T_1, T_2, are fixed together in one piece, and placed, when in use, in a metal frame which slides vertically, either upon a wire passing through it and provided with a plummet which rests upon the bottom of the stream to keep the wire stretched, or the tubes, T_1, T_2, may be fixed upon a vertical wooden pole with its lower end driven into the bed of the stream. In either case, accurate means is provided for showing the depth to which the instrument is submerged.

By making the vertical gauge scale adjustable and placing it with its zero opposite the top of the lower column at each change of depth of instrument, the necessity of observing the height of both columns at each reading is obviated, since the reading of the upper column alone gives the head, h, at once.

The Ferris-Pitot Meter.—This meter, which was invented by Mr. Walter Ferris, of Philadelphia, is designed to measure the flow of liquids in pipes when quite full. It consists of a device for the registration of the results obtained by the Pitot tube. Also, special devices are fitted to it to prevent the clogging of the tubes and to permit of their examination when in use.

In the following sketch, if P represents the level at which the water stands in the straight Pitot tube, T_1, then $h = kv^2$. Or, the difference in level between the columns in the two tubes, T_1 and T_2, is the head, h, due to the velocity of the water in the pipe as it impinges against the open up-stream end of the bent tube, T_2. For a given velocity, v, this difference, h, is constant, and is independent of the pressure represented by P.

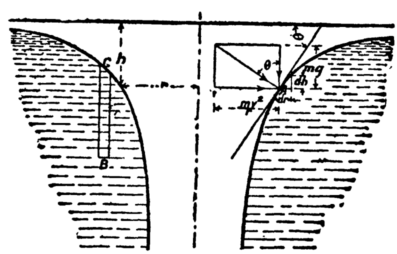
The Ferris register, like the Venturi Meter, records the velocity (existing at the instant of registration) in terms of the total discharge since the last registry, and as an increase in the total number of cubic feet registered. The registry thus involves the assumption that the average velocity, during the

period between registrations, is equal to the velocity at the end of that period.

This meter is usually calibrated by reference to some accepted standard, and the coefficient or coefficients thus obtained are used in subsequent observations, while the registration of the meter is made every two minutes. Careful experiments proved that this meter is correct under ordinary circumstances to within 3 per cent.

The principle of the Pitot tube has been employed also to measure and record the speed of ships. Two tubes are provided: one pointing in the direction of the motion of the ship, and the other pointing in the opposite direction. These tubes are connected up to a differential pressure gauge of the elastic diaphragm or other type, and the indication of the pressure gauge is communicated by a special mechanism to a recording pencil which gives upon a time-driven sheet a graph of the speeds at different times.

Vortex Motion.—A whirling mass of fluid is termed a *vortex* and may be either *forced* or *free*. A *free vortex* is one which can be formed naturally, as when water flows through a hole at the bottom of a basin, and is such, that the energy of the fluid per unit mass is the same at all points in it. A *forced vortex* can only be produced artificially, and in it the energy of the fluid is different at different places. Such a vortex may be obtained by rotating a cup containing water.



FREE VORTEX.

Free Vortex.—From the above condition of constant energy, the velocity at a point A, on the surface of the vortex and at a depth h below the level of the still water, must be the same as that due to a body falling freely from a height h.

$$\therefore \qquad v = \sqrt{2 g h}.$$

A particle on the surface of the vortex is acted upon vertically by its own weight $m\,g$, and horizontally by a centrifugal force $\frac{m\,v^2}{r}$. The resultant of these two forces must make an angle θ, with the vertical, so that:—

$$\tan\theta = \frac{\frac{m\,v^2}{r}}{m\,g} = \frac{v^2}{g\,r} = \frac{2\,h}{r}.$$

Here m is the mass of the particle and r the radius of the circle in which it revolves.

This resultant must also be perpendicular to the surface of the fluid at A, since the fluid cannot resist any shearing component. Consequently, if the slope of the surface at A be $\tan\theta$,

$$-\frac{d\,h}{d\,r} = \tan\theta = \frac{2\,h}{r}.$$

Or,
$$\frac{d\,h}{2\,h} = -\frac{d\,r}{r},$$

i.e.,
$$\tfrac{1}{2}\log h = -\log r + \log c,$$

$\therefore$
$$\log\sqrt{h} = \log\frac{c}{r},$$

$\therefore$
$$h = \frac{c^2}{r^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(XIII)}$$

Here, c is a constant of integration which will depend on the size of the vortex. It is the radius of the vortex at unit depth.

Since the total energy is the same at all places in the fluid and the pressure at B due to the column B C, just makes up for the lower level of B, the velocity in any thin vertical tube of fluid such as B C must be constant and equal to that at the surface of the tube.

A vortex of this kind was applied to centrifugal pumps with radial blades by Professor James Thomson in order to convert, as far as possible, the kinetic energy of the water into potential energy. The vortex commences at the circumference of the wheel and the radial component of the water's motion is outward.

Forced Vortex.—In a vortex whose angular velocity ω is constant throughout, we have at any point A on the surface :—

$$\text{Centrifugal force} = m\,r\,\omega^2,$$

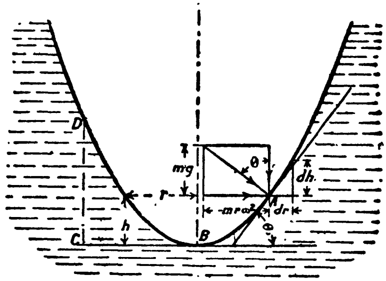
$\therefore$
$$\tan\theta = \frac{m\,r\,\omega^2}{m\,g} = \frac{r\,\omega^2}{g}.$$

$\therefore$
$$\frac{d\,h}{d\,r} = \frac{\omega^2}{g}\,r.$$

Or,
$$d\,h = \frac{\omega^2}{g}\,r\,d\,r.$$

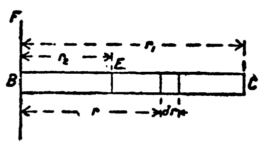
$\therefore$
$$h = \frac{\omega^2\,r^2}{2\,g} = \frac{v^2}{2\,g}. \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{(XIV)}$$

This is the equation to a parabola with its axis vertical, and, therefore, the surface of the vortex will be the paraboloid formed by rotating this parabola about its axis. The velocity of any particle D, on the surface of the vortex is that which would be attained by a body falling a distance equal to D C. The kinetic and potential energies are both least at the centre B, and become greater as we move upwards or outwards. We have a vortex of this kind in the wheel of a centrifugal pump with radial blades.



FORCED VORTEX.

Pressure due to Centrifugal Force.—When a liquid is rotating, its pressure is not the same for all points on one level. Thus, in the previous figure the pressure at B is atmospheric, while at the point C on the same level, the pressure is, in addition, that due to the head C D.



WHIRLING COLUMN.

Consider a uniform column of liquid B C, rotating round the axis B F, with an angular velocity ω and cross section a. Then, if ρ be the density of the liquid, the centrifugal force due to a small element of length dr, at a distance r from the axis is :—

$$\rho a \, dr \times \omega^2 r.$$

$\therefore$ The total centrifugal force of the column B C $= \rho a \omega^2 \int r \, dr$.

This force is spread over an area a, at the end C, and we must divide it by this area to get p, the pressure per unit area. If the whole column from B to C is full of liquid :—

Then, $$p = \rho \omega^2 \int_0^{r_1} r \, dr = \tfrac{1}{2} \rho \omega^2 r_1^2 = \tfrac{1}{2} \rho v_1^2 \quad . \quad . \quad . \quad \text{(XV)}$$

But, if the part of the column from B to E is empty :—

Then, $$p = \rho \omega^2 \int_{r_2}^{r_1} r \, dr = \tfrac{1}{2} \rho \omega^2 (r_1^2 - r_2^2) \quad . \quad . \quad \text{(XV}_a\text{)}$$

Students who are not acquainted with the integral calculus will understand the following proof of these equations.

The centre of gravity of B C is at a distance of $\frac{1}{2}$ B C from B.

$$\therefore \left.\begin{array}{l}\text{Centrifugal force due to}\\ \text{the whole column B C}\end{array}\right\} = \rho\, a\, r_1 \times \omega^2 \frac{r_1}{2} = \tfrac{1}{2}\rho\, a\, \omega^2 r_1^2.$$

The centre of gravity of the part E C is distant $\frac{r_1 + r_2}{2}$ from B.

$$\therefore \text{ Centrifugal force due to E C} = \rho\, a\,(r_1 - r_2)\,\omega^2 \frac{r_1 + r_2}{2}$$

$$\text{,,} \qquad \text{,,} \qquad = \tfrac{1}{2}\rho\, a\, \omega^2 (r_1^2 - r_2^2).$$

This formula may be applied to finding the equation for the forced vortex. For, in the figure of the *forced vortex*, if B C = r, and C D = h, then, $p = h\,w = h\,\rho\,g$.

Therefore, from equation (XV), we get :—

$$h\,\rho\,g = \tfrac{1}{2}\rho\, v^2,$$

Or,

$$h = \frac{v^2}{2g} \text{ (as before).}$$

Reaction of a Jet.—In general, when a fluid issues from an orifice it exerts a force on the vessel which contained it. This force is the reaction of the jet, and is due to the momentum given to the escaping fluid.

Let v = Velocity of efflux, and a = cross-sectional area of outlet.
,, w = Weight of unit volume of the fluid.
,, W = Weight of fluid issuing per second.
,, m = Mass issuing per second.
,, Q = Quantity or volume issuing per second.
,, F = Force exerted on the vessel containing the fluid.

Then, the momentum given to the water per second = $m\,v$. And, since force is the rate of change of momentum :—

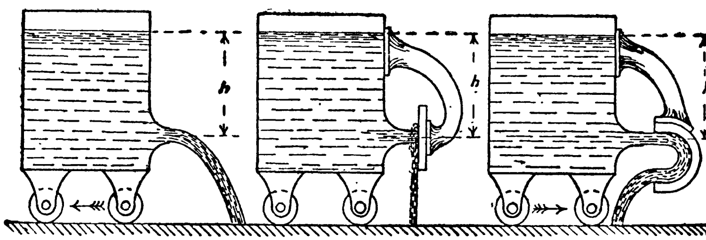
$$F = m\,v = \frac{W}{g}\,v.$$

But, $\quad W = w\,Q = w\,a\,v.$

$$\therefore \qquad F = \frac{W}{g}\,v = \frac{w\,a\,v^2}{g} = \frac{w\,a\,.\,2\,g\,h}{g} = 2\,w\,a\,h. \qquad \text{(XVI)}$$

This formula also gives the force acting on a vane of a turbine or waterwheel which alters the direction of flow of the water, and

in that case, v is the change of velocity produced by the vane. In applying the formula to any particular case, v is the total change of velocity produced until the water is quite clear of the vessel, and the direction of the force acting on the vessel is exactly opposite to that of v. The following figures will explain the results in three cases :—



BACKWARD. NO MOTION. FORWARD.

MOTION PRODUCED BY A JET.

In the left-hand figure the jet of liquid issues towards the right and urges the containing vessel to the left. The jet in the central figure strikes a plate fixed to the tank close to it, and then drops vertically downwards; consequently, the water receives no horizontal momentum and the tank no motion from it. The water will, however, exert a pressure on the plate, and this pressure balances the force produced by issuing from the orifice. In the remaining figure the jet is turned backwards by a curved guide attached to the tank and the whole momentum imparted to the water is backwards, consequently the tank is pressed forwards. If there be no loss from friction or eddies, the backward velocity of the stream, when the tank is at rest, must be the same as that with which it left the orifice. The blade has not only stopped the water, but has given it an equal backward momentum, and must, therefore, be pressed with a force twice as great as the flat one in the previous case. The force on it, is therefore, 2 F or 4 $w\,a\,h$, which is four times the pressure on a plug stopping the orifice. When the tank is in motion, the momentum given to the water is modified thereby, as will be explained in connection with the Pelton wheel in the next Lecture. Steam life-boats are now frequently propelled by jets of water instead of by screw propellers.

As this reaction astonishes everyone who hears of it for the first time, we will consider it from another point of view. Water under a pressure of 100 lbs. per square inch issues out of a nozzle of 1 square inch in area, and impinges on a fixed vane of

such a shape as to gradually deflect the water and turn it back in the opposite direction, without loss by friction, and consequently without loss of velocity. Thus, the vane may be a double U, as in the Pelton wheel; or a semicircle; or it may deflect the water in more than one plane. Then, the pressure on the vane is 400 lbs., although the statical pressure on a plug or valve stopping the jet is only 100 lbs. Although it requires very little elementary dynamics to prove this fact, the statement is generally received with incredulity.

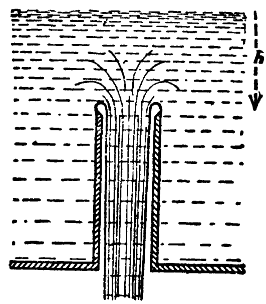
In the first place, it will be seen that the vane not only arrests the water—that is, imparts a negative acceleration to it, equal and opposite to what it received from the nozzle—but accelerates it equally in the opposite direction. Therefore, the total pressure on the vane is double the reaction on the nozzle.

The nozzle reaction at first sight appears to be 100 lbs.; that is, area of jet multiplied by pressure. But wait! Imagine a square vertical column in which water is maintained at a constant head. Near the bottom is a suitably formed horizontal nozzle closed by a plug. The system is then in equilibrium, since every square inch of wall has another square inch opposite to it which is acted on by an equal and opposite pressure. Opposite the nozzle, on the other side of the vessel, is a small circle equal to the area of the plug and equally pressed by the water. Now, remove the plug and let the water issue. It really does look as if the force pushing the vessel back was the pressure due to the head acting on this small circle on the back wall—that is, equal to the pressure multiplied by area of jet. But by Newton's second law, whenever we find a body moving at any velocity, we know the product of the force which has urged it and the time during which it has acted. Let us take a simple case and suppose the height of the water surface above the nozzle to be 16 feet. Then we know that the water issues as fast as if it had fallen 16 feet, and that its velocity is 32 feet per second. Consequently, every second a cylinder of water equal in section to the area of the jet and 32 feet long has a velocity of 32 feet per second impressed on it. Therefore, the urging force must be equal to its own weight; because when falling from rest by gravity—that is, when urged by its own weight—it acquires that velocity in one second. But the statical pressure is only half of this, being the weight of a cylinder of water equal in section to the jet and 16 feet long; and, since action and reaction are equal and opposite, the recoil of the vessel is equal to the force urging the jet, or twice the statical pressure.

Of course, if the jet strikes a flat vane at right angles the motion is stopped in its own direction, but not returned, and the pressure on the vane is twice the statical pressure. But, if the vane

returns the motion undiminished the pressure is four times the statical pressure.

Reduction of Pressure Round an Orifice.—Let h be the depth of a bell-shaped orifice and a its cross sectional area; then, if this orifice be completely closed by a flat plate, the force required to keep the plate in position will be $w\,a\,h$. It might therefore be supposed that, when the plate is removed and the water allowed to flow, this would be the force exerted on the vessel, but on looking at our former result (see equation XVI) we find that the actual force is twice as great, or $2\,w\,a\,h$. This difference is caused by a diminution of pressure on the surface round about the orifice. At these places the water has a certain velocity, and Bernouilli's theorem shows us that the pressure there must be less than when the liquid was at rest.



RE-ENTRANT ORIFICE.

There can be no reduction of pressure on the flat surface round the bottom of a re-entrant orifice, because there the water is practically at rest. The amount by which the pressure is reduced on opening the orifice must, therefore, be that on the area of the orifice itself, or $w\,a\,h$. Every second, this will set in motion a mass m as given by the equation :—

$$w\,a\,h = m\,v.$$

But, $$m = \frac{W}{g} = \frac{w\,Q}{g} = \frac{w\,a'\,v}{g}.$$

Where a' is the cross area of the jet, and $v = \sqrt{2\,g\,h}$.

$$\therefore \qquad w\,a\,h = \frac{w\,a'\,v^2}{g} = \frac{w\,a' \times 2\,g\,h}{g}.$$

Hence, $$a = 2\,a'. \qquad \text{(XVII)}$$

That is, the area of the jet will only be half that of the orifice. This result has been found experimentally to be correct. With a sharp-edged orifice, such as we considered in the early part of this lecture, there is a certain reduction of pressure round the orifice, and, therefore, the contraction must be less than in this case.

Impact—Loss of Energy.—When a stream of fluid meets an obstruction which causes a sudden change in its motion, its kinetic energy is partially, or perhaps wholly, spent in forming eddies or

little whirls of water, and is thus lost so far as useful effects are concerned. The eddies are, however, soon stilled by the viscosity of the liquid and their energy is converted into heat.

Let us suppose a body of mass m_1 moving with a velocity v_1, to overtake and strike another body of mass m_2, and velocity v_2. After collision let the bodies move on together with a common velocity v.

Then, from the laws of momentum,

$$(m_1 + m_2)\, v = m_1 v_1 + m_2 v_2.$$

Or,
$$v = \frac{m_1 v_1 + m_2 v_2}{m_1 + m_2}.$$

But, before impact, the total energy was :—

$$E_1 = m_1 v_1^2 + m_2 v_2^2$$

And, after impact :—

$$E_2 = (m_1 + m_2)\, v^2.$$

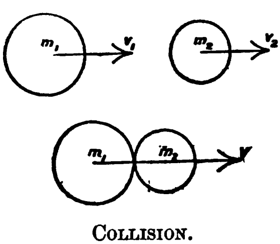
$$\therefore \quad E_2 = (m_1 + m_2)\left\{\frac{m_1 v_1 + m_2 v_2}{m_1 + m_2}\right\}^2 = \frac{(m_1 v_1 + m_2 v_2)^2}{m_1 + m_2}.$$

The energy lost is :—

$$E_1 - E_2 = \frac{(m_1 v_1^2 + m_2 v_2^2)(m_1 + m_2) - (m_1 v_1 + m_2 v_2)^2}{m_1 + m_2}$$

$$\therefore \; E_1 - E_2 = \frac{m_1 m_2 (v_1^2 + v_2^2 - 2 v_1 v_2)}{m_1 + m_2} = \frac{m_1 m_2}{m_1 + m_2}(v_1 - v_2)^2. \text{(XVIII)}$$

This lost energy is converted into heat. If one or both of the bodies be fluids the lost energy at first shows itself in eddies; but, as already explained, it is ultimately converted into heat. We thus see that when two streams moving at different velocities mix together, energy is lost and this loss is greater the greater the difference of the velocities. A similar effect takes place when water flows through a pipe with sudden changes of area, and even to a slight extent when the area is gradually varied, and also when water is flowing in a pipe or channel above a certain speed. In all these cases, different parts of the water move with



COLLISION.

different velocities, and these parts get mixed with one another. It also shows, why a turbine or water-wheel in which the water collides with the vanes, must have a low efficiency.

Resistance of a Pipe.—When a fluid is passing through a pipe it rubs against the sides and experiences a certain resistance to its motion. This resistance limits the flow in a long pipe and causes a loss of head or pressure. Had the water no viscosity its flow would not be affected by the friction of the inner surface; because, this friction could only act on the thin layer of water actually in contact with the pipe.

Professor Reynolds found that the manner in which water flows depends upon its velocity. When this is below a certain critical point the flow depends chiefly on the viscosity and is along smooth stream lines. On passing the critical velocity, the water no longer moves steadily, but breaks up into numberless little whirls or eddies which move along with it, and absorb energy from the main stream. The former condition may be called *Steady Flow* and the latter *Eddy Flow*. This was beautifully demonstrated by Professor Reynolds by passing water from a large tank through a glass tube and then injecting a fine stream of coloured liquid with the same velocity into the centre of the stream. The water in the tank was quite steady and entered the tube through a bell mouth. At low velocities the coloured liquid formed a thin line along the centre of the tube, but at a certain velocity it was seen to suddenly spread out through a considerable part of the water, and on photographing it by means of an electric spark it was found to be all twisted into little whirls.

RESISTANCE OF SMOOTH LEAD PIPE AT DIFFERENT SPEEDS OF A FLUID.

He also found, that the law connecting the resistance R, with the velocity v, was different for these two conditions. For the lower speeds he ascertained that the resistance was proportional to the velocity, but at greater speeds it varied as some higher power, ranging from 1·7 to 2 depending on the roughness of the pipe. Near the critical velocity, which depends on the diameter of the pipe and on the temperature, the law is uncertain. The temperature affects the viscosity on which the critical velocity also depends.

On plotting the logarithms of his results as obtained with a smooth lead pipe $\frac{1}{2}$ inch in diameter, we get the lines shown in the accompanying figure. This consists of two straight lines joined by a

curve of indefinite shape. The two straight parts have slopes of 1 and 1·72, showing that in one case the increase of log R is equal to the corresponding increment of log v, whereas beyond a certain point it is 1·72 times the corresponding increment of log v.

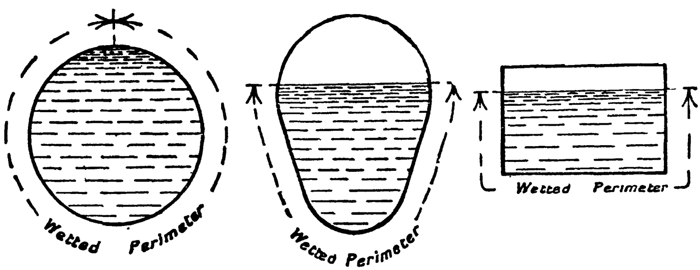
This shows us, that if c_1, c_2, and c_3 be constants, then :—

$$\text{For steady flow, } R = c_1 v.$$

$$\text{For eddy flow, } R = c_2 v^{1\cdot72} + c_3.$$

The resistance of the pipe for any velocity is most conveniently expressed in terms of the difference of pressure per unit length required to force the necessary quantity of liquid per second through it when level. This is called the *Slope of Pressure.* Whether the pipe is level or not, if it be of uniform bore, this slope is given by the difference of level between the free surfaces of the pressure columns at any two points, divided by the length of pipe between them. This has been termed the *Slope of Free Level,* and is shown graphically by a line passing through the surfaces of a series of little pressure columns.

Hydraulic Mean Depth.—The resistance of a pipe or channel is directly proportional to the extent of the wetted surface in a given



WETTED PERIMETER OF DIFFERENT SECTIONS.

length, and inversely to the cross area of the stream. The transverse length of the whole surface wetted by the stream is called the *Wetted Perimeter*, and the ratio of the cross area of the stream to its wetted perimeter is termed the *Hydraulic Mean Depth*, which we shall denote by D.

The friction of the pipe or channel will consequently vary inversely as the hydraulic mean depth.

Combining this with the former results we have for water pipes or channels :—

$$R = c\frac{v^n}{D}. \qquad \ldots \ldots \quad \text{(XIX)}$$

Where c is a constant depending on the nature of the surface and the temperature of the water, and n is a number varying from 1·7 for smooth lead pipes to 2 for very rough pipes. The critical velocity for water in ordinary pipes is so low, that in all practical cases in which we wish to know the resistance, the actual velocity is always above it.

For a circular pipe, whose diameter is d, and which is full of running water, the wetted perimeter is the circumference of the circle :—

$$\therefore \qquad D = \frac{\frac{\pi}{4} d^2}{\pi d} = \frac{d}{4} \text{ for a circular pipe.} \qquad \text{(XX)}$$

For a rectangular open stream of depth h and breadth b, the wetted perimeter $= 2h + b$ and the area is $b\,h$:—

$$\therefore \qquad D = \frac{b\,h}{2h + b} \text{ for an open rectangle.} \qquad \text{(XXI)}$$

Loss of Head due to Friction in Water Pipes.*—Having dealt with the loss of "head" where water enters a pipe due to the shape of its mouth, and mentioned Prof. Reynold's experiments on steady and eddy flows of the liquid through a smooth pipe, as well as explained what is meant by the "wetted perimeter," we are in a position to state the generally accepted rules for the loss of head in ordinary clean cast-iron water pipes.

From many tests with piezometers or pressure columns, gauges or vertical tubes, inserted at known distances apart into the upper sides of straight pipes of different sizes and internal roughness, but of uniform bore, without bends, it has been found that *the loss of head due to friction* for water flowing through these pipes at different velocities, is approximately proportional to—

1. The length of the pipe, l.
2. Inversely to the diameter of the pipe, d.
3. The square of the velocity, v.
4. The roughness.
5. But it is independent of the water pressure.

Let h_T = Total head in feet or pressure per sq. in.
h_L = Lost head due to friction " " "
h_E = Effective head at point of delivery " "

Then, $$h_L = h_T - h_E$$

* This "loss of head due to friction" is termed the "friction head."

Now, if f = coefficient of friction found by experiment, which will vary between 0·012 for a $\frac{1}{2}$-inch pipe when v = 1 foot per second, and 0·003 for a 3-foot pipe when v = 15 feet per second.

Then, loss of head,

$$h_L = f\left(\frac{\text{wetted surface}}{\text{cross area}} \times \text{loss of head due to velocity}\right).$$

$$\text{Or,}\quad h_L = f\left(\frac{\pi\, d\, l}{\frac{\pi}{4}d^2} \times \frac{v^2}{2\,g}\right)$$

$$\therefore\quad h_L = f\left(\frac{4\,l\,v^2}{d\,2\,g}\right). \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad \text{(XXII)}$$

EXAMPLE II.—What will be the loss in head for every 100 feet of a 3-inch pipe when water flows through it with a velocity of 3 feet per second, if the coefficient of friction be ·0065?

Here, l = 100 feet; d = 3 inches = ·25 feet; v = 3 feet per second; and f = ·0065.

By formula XXII., ANSWER—

We see that, $$h_L = f\left(\frac{4\,l\,v^2}{d\,2\,g}\right).$$

$$\therefore\quad h_L = \cdot0065\left(\frac{4 \times 100 \times 3 \times 3}{\cdot25 \times 2 \times 32\cdot2}\right).$$

Or, $$h_L = 1\cdot46 \text{ feet.}$$

Now, looking at the following table by Prof. Merriman, we see that opposite to a pipe of 0·25 foot in diameter, and directly under the velocity 3 feet per second, the loss of head is printed as 1·46 feet for every 100 feet of length. In that table, the "friction factor," f_F, is reckoned as four times the previously mentioned value for the coefficient of friction, f.

Or, $$f_F = 4\,f;$$

because the constant 4 appears in the numerator of the natural equation XXII.

Hence, when using the table, the "friction factors," f_F, will vary from 0·05 for a $\frac{1}{2}$-inch pipe when v = 1 foot per second, to 0·012 for a 3-foot pipe when v = 15 feet per second.

The formula for the "friction head" therefore becomes, using the following table:—

$$h_L = 4\,f\left(\frac{l\,v^2}{2\,d\,g}\right) = f_F\left(\frac{l\,v^2}{2\,d\,g}\right). \quad . \quad . \quad . \qquad \text{(XXIII)}$$

TABLE OF FRICTION HEAD FOR 100 FEET OF CLEAN IRON PIPE.*

Diameter of Pipe.	Velocity in Feet per Second.						
	1.	2.	3.	4.	6.	10.	15.
Feet.	Feet.	Feet.	Feet.	Feet.	Feet.	Feet.	Feet.
0·05	1·46	5·10	10·30	16·90	34·70	...	...
0·10	0·59	1·99	4·20	6·97	14·50	37·30	...
0·25	0·20	0·70	1·46	2·40	5·37	13·70	29·40
0·50	0·09	0·32	0·70	1·14	2·46	6·22	13·30
0·75	0·05	0·21	0·45	0·73	1·57	3·94	8·40
1·00	0·04	0·15	0·32	0·55	1·12	2·80	5·95
1·25	0·03	0·11	0·25	0·42	0·85	2·11	4·48
1·50	0·02	0·09	0·20	0·33	0·67	1·66	3·50
1·75	0·02	0·07	0·16	0·26	0·54	1·33	2·80
2·00	0·02	0·06	0·13	0·21	0·45	1·09	2·27
2·50	0·01	0·05	0·10	0·16	0·34	0·81	1·68
3·00	0·01	0·04	0·07	0·12	0·26	0·67	1·40
3·50	0·01	0·03	0·06	0·10	0·21	0·53	...
4·00	...	0·02	0·05	0·08	0·17	0·42	...
5·00	...	0·02	0·04	0·06	0·13	...	...
6·00	...	0·01	0·03	0·05	0·10	...	...

EXAMPLE III.—Find the friction in a 4-inch wrought-iron pipe delivering 200 gallons per minute. What is the loss in head and H.P. per 100 feet? (See *The Practical Engineer*, Sept. 11, 1903, p. 264.)

ANSWER.—Gallons per second $= \frac{200}{60}$.

Cubic feet per second $= \frac{200 \times 10}{60 \times 62{\cdot}3} = {\cdot}535.$

Area in sq. ft. of a 4″ pipe $= {\cdot}087.$

Velocity (v) of water $= \frac{{\cdot}535}{{\cdot}087} = 6{\cdot}1$ ft. per sec.

Let the friction factor $= {\cdot}023.$

Then, by formula XXIII. and the above table—

We get, $$h_L = f_F\left(\frac{l\,v^2}{2\,d\,g}\right) = {\cdot}023\left(\frac{100 \times 6{\cdot}1^2}{2 \times {\cdot}\dot{3} \times 32{\cdot}2}\right) = 4 \text{ ft.}$$

H.P. absorbed $= \frac{4 \times 200 \times 10}{33{,}000} = {\cdot}24$ (approximately).

* From *Treatise on Hydraulics*, 8th Edition, by Prof. Merriman, Lehigh University

Example IV.—A hydraulic lift, with ram, load, &c., weighs 10 tons, the ram is 9 inches in diameter, and the friction in the mechanism is equal to 0·05 of the gross load. The accumulator is half a mile away, and is loaded to 800 lbs. per square inch, the diameter of the supply pipe is 3 inches. Estimate the speed of ascent of the lift, if the loss of head in the pipe is $0{\cdot}0005\, l v^2/d$; l being the length in feet, d the diameter in feet, v the velocity of water in the pipe in feet per second. (B. of E., S. 3, 1904.)

Answer.—Let V be the speed of the lift in feet per second. Then, since the diameter of the supply pipe is $\frac{1}{3}$ the diameter of the ram, the speed of water in the supply pipe will be equal to 9 V. Assuming that the "loss of head" is measured in feet of water, we get:—

$$\left.\begin{array}{r}\text{The loss of pressure in}\\ \text{lbs. per square inch}\end{array}\right\} = {\cdot}43 \times \frac{0{\cdot}0005\, l v^2}{d} \left\{\begin{array}{l}\text{Since } {\cdot}43 \text{ lb. per}\\ \text{sq. in.} = 1 \text{ foot}\\ \text{head of water.}\end{array}\right.$$

$$\text{,,} \qquad \text{,,} \qquad = \frac{{\cdot}43 \times 0{\cdot}0005 \times 2{,}640 \times (9\text{ V})^2}{{\cdot}25}$$

$$\text{,,} \qquad \text{,,} \qquad = 184\text{ V}^2.$$

Thus, the pressure on the ram will be equal to $(800 - 184\text{ V}^2)$ lbs. per square inch.

Equating the work done per second by the water on the ram to the work per second required to raise the lift, &c., we have—

$$\left.\begin{array}{l}\text{Pressure per sq. inch}\\ \text{on the ram} \times \text{area of}\\ \text{ram} \times \text{its speed}\end{array}\right\} = \left\{\begin{array}{l}\text{(Weight of lift + loss due to its}\\ \text{friction)} \times \text{its speed.}\end{array}\right.$$

$$(800 - 184\text{ V}^2) \times \frac{\pi \times 9^2}{4} \times \text{V} = (10 \times 2{,}240 + {\cdot}05 \times 10 \times 2{,}240) \times \text{V}.$$

$$\text{,,} \qquad \text{,,} \qquad = 10 \times 2{,}240(1 + {\cdot}05) \times \text{V}.$$

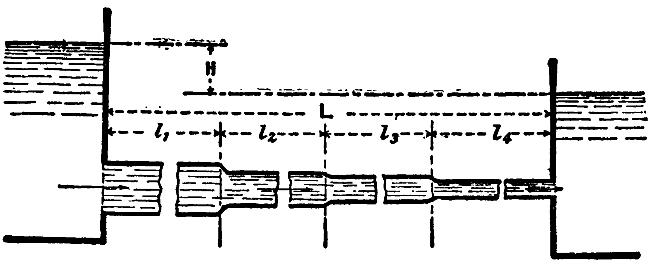
Hence,
$$\text{V}^2 = \frac{800}{184} - \frac{10 \times 2{,}240 \times 1{\cdot}05 \times 4}{\pi \times 9^2 \times 184}.$$

Or,
$$\text{V}^2 = 4{\cdot}35 - 2{\cdot}01 = 2{\cdot}34.$$

$$\therefore \qquad \text{V} = \sqrt{2{\cdot}34} = 1{\cdot}53 \text{ feet per second.}$$

To find the discharge through a long pipe line consisting of a number of pipes of different diameters.

If it is desirable to diminish the diameter of a long pipe line, instead of using a pipe the diameter of which varies uniformly with the length, then the line is made up of a number of parallel pipes of different diameters and lengths, since only short conical pipes are used in practice, as, for instance, in the limbs of a Venturi water meter.



THE DISCHARGE THROUGH A LONG PIPE CONSISTING OF A NUMBER OF PIPES OF DIFFERENT DIAMETERS.

Let $l_1, l_2, l_3, l_4 \ldots$ be the lengths of the several portions of the pipe;

$d_1, d_2, d_3, d_4 \ldots$ be the respective diameters of the sections of the pipe;

$v_1, v_2, v_3, v_4 \ldots$ be the corresponding velocities;

$F_1, F_2, F_3, F_4, \ldots$ be the corresponding values of the resistance or "friction" factor $= 4f$;

$L = l_1 + l_2 + l_3 + l_4 =$ the total length of pipe;

$H =$ the total head;

$Q =$ the rate of discharge $= \frac{\pi}{4} d_1{}^2 v_1 = \frac{\pi}{4} d_2{}^2 v_2 =$ &c., &c.;

$H =$ total head $=$ friction head, since, in a long pipe, the velocity and entry heads are usually negligible relatively to the friction head, and are neglected.

In each portion of the pipe, the resistance and the corresponding "friction" head, h_L, are believed to be proportional directly to the length, l, of such portion and to the velocity head, $\frac{v^2}{2g}$, and inversely to the diameter, d; or

$h_L = F \times \frac{l}{d} \times \frac{v^2}{2g}$. (See Vol. IV., Lecture IV., Equation XXII.)

Hence,

$$H = F_1 \frac{l_1}{d_1} \times \frac{v_1^2}{2g} + F_2 \frac{l_2}{d_2} \times \frac{v_2^2}{2g} + F_3 \frac{l_3}{d_3} \times \frac{v_3^2}{2g} + F_4 \frac{l_4}{d_4} \times \frac{v_4^2}{2g} + \ldots \&c.,$$

and since $v_1 = \frac{4Q}{\pi d_1^2}$, $v_2 = \frac{4Q}{\pi d_2^2}$, &c., we have also

$$H = F_1 \frac{l_1}{d_1} \times \frac{16 Q^2}{2 g \pi^2 d_1^4} + F_2 \frac{l_2}{d_2} \times \frac{16 Q^2}{2 g \pi^2 d_2^4} + F_3 \frac{l_3}{d_3} \times \frac{16 Q^2}{2 g \pi^2 d_3^4}$$
$$+ F_4 \frac{l_4}{d_4} \times \frac{16 Q^2}{2 g \pi^2 d_4^4} \times \&c.$$

$$H = \frac{16 Q^2}{\pi^2 2 g}\left(F_1 \frac{l_1}{d_1^5} + F_2 \frac{l_2}{d_2^5} + F_3 \frac{l_3}{d_3^5} + F_4 \frac{l_4}{d_4^5} + \&c.\right),$$

or,

$$H = \frac{64 Q^2}{\pi^2 2 g}\left(f_1 \frac{l_1}{d_1^5} + f_2 \frac{l_2}{d_2^5} + f_3 \frac{l_3}{d_3^5} + f_4 \frac{l_4}{d_4^5} + \&c.\right),$$

whence,

$$Q = \frac{\pi}{8}\sqrt{\frac{2gH}{f_1 \frac{l_1}{d_1^5} + f_2 \frac{l_2}{d_2^5} + f_3 \frac{l_3}{d_3^5} + f_4 \frac{l_4}{d_4^5} + \&c.}}$$

In approximate calculations of the head lost in such mains, it is generally accurate enough to neglect the smaller losses of head and to have regard to the pipe friction only, and then the calculations may be facilitated by reducing the main to a main of uniform diameter, in which there would be the same loss of head. Such a uniform main is generally termed *an equivalent main.*

Suppose that f may be treated as constant for all the pipes.

Then, the diameter, d, of the pipe, which, for the same total length, would give the same discharge for the same loss of head due to friction, can be found from the equation—

$$\frac{L}{d^5} = \frac{l_1 + l_2 + l_3 + l_4}{d^5} = \frac{l_1}{d_1^5} + \frac{l_2}{d_2^5} + \frac{l_3}{d_3^5} + \frac{l_4}{d_4^5} + \ldots$$

The length, L_1, of the equivalent uniform main, of constant

diameter, D, which will give the same discharge for the same loss of head by friction, is—

$$L_1 = \left(\frac{D}{d_1}\right)^5 l_1 + \left(\frac{D}{d_2}\right)^5 l_2 + \left(\frac{D}{d_3}\right)^5 l_3 + \left(\frac{D}{d_4}\right)^5 l_4 + \text{\&c.}$$

Also, the length of equivalent uniform main, which would have the same total loss of head for any given discharge, is—

$$L_2 = \left(\frac{d}{d_1}\right)^5 l_1 + \left(\frac{d}{d_2}\right)^5 l_2 + \left(\frac{d}{d_3}\right)^5 l_3 + \left(\frac{d}{d_4}\right)^5 l_4 + \text{\&c.}$$

Lecture IV.—Questions.

1. Prove the law for changing p, v, and h along a stream line in a frictionless fluid. Apply the law to find the funnel shape of the surface of water in a basin from which the water is flowing by a central hole.

2. Prove the law for changing p, v, and h along the stream lines in a frictionless fluid. Apply, neglecting change of level, to the case of adiabatic flow of air from one vessel to another through a small orifice, and deduce the rule for maximum quantity flowing.

3. Describe the action of a jet pump, or of a good form of injector.

4. Water flows through a round sharp-edged orifice 3 inches in diameter in a flat plate, at about 12 inches below still-water level. Show by a sketch your notion of the shapes of the stream lines. If we wish to know the pressure at any point, why is it not sufficient to know only the depth?

5. Deduce a formula giving the velocity with which water issues from an orifice, and show how to apply it to water under pressure.

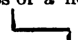
6. Discuss briefly the relative advantages under various circumstances of the different methods of measuring a stream of water.

7. Find the horse-power of a waterfall 70 feet high, when the stream is such that it passes over a gauge notch 6 feet long with a head of 15 inches. If it is employed to drive a turbine of 80 per cent. efficiency, what B.H.P. would you expect to obtain? *Ans.* 184·5 B.H.P.

8. Investigate the form of the surface of water which flows out of a hole in the bottom of a basin with a vortex motion.

9. What is meant by a constrained vortex, and why will such a vortex rapidly disappear when left to itself? Find the form taken by the surface and show how to apply this to finding the pressure in a centrifugal pump.

10. The wheel of a centrifugal pump 2 feet outside diameter has a very large case and rotates at 100 revolutions per minute about a vertical axis, and almost no water is being delivered. Calculate and show in a curve the pressure at points in a horizontal plane, at various distances from the axis. The vanes are bent backwards at an angle of 30° with the radius at the outer part; if the radial flow becomes 2 feet per second and the circumferential openings are 200 sq. inches in area, what is the kinetic energy of the water leaving the wheel? What is the pressure in excess of that at the inner rim of the wheel where the water enters without shock? If there is no frictional loss to what height will the water be lifted above the well? In an actual pump with small wheel case what is the probable lift?

11. "Barker's mill" consists of a horizontal pipe with a nozzle at right angles to it at each end thus  Water enters it by a vertical pipe at the centre, and the whole is so mounted that it can rotate about the axis of this pipe. Find the torque when the water is issuing under a head h, and the mill is revolving n times per minute. Show that the power is greatest when the velocity of the nozzles is half that due to the head, and that the efficiency then cannot be over 50 per cent.

12. What are the chief conclusions to be drawn from Reynold's experiments on the flow of water through pipes?

13. What is meant by the *Hydraulic Mean Depth* of a pipe or channel, and show how we use it in calculating the resistance to the flow of water?

14. If a hydraulic company supplies 1,000 gallons of water at 700 lbs. pressure per square inch for 17 pence, how much is this per horse-power per hour? Another company supplies water at a "head" of 260 feet: what price ought it to charge per 1,000 gallons if the consumer is to have his energy at the same price as in the other case? *Ans.* 2·08d.; $2\frac{3}{4}$d.

15. What is the usually accepted rule for loss of head or energy per pound of water passing along a straight pipe? What is the law when the water flows very slowly? And why is there a difference between the two laws? (B. of E. Adv., 1900.)

16. 60 horse-power is being delivered hydraulically into a mile of straight horizontal pipe at a pressure of 700 lbs. per square inch. It is found that 52 horse-power is available at the far end. What power will be found available at the end of 2 miles of a pipe three-fourths of the diameter of the first-named, if the entering pressure is 800 lbs. per square inch, and the entering horse-power is 40? *Ans.* 26·61 H.P.

17. A horizontal pipe of 12 inches diameter gradually becomes 3 inches diameter, and then becomes of 12 inches diameter again. There is a flow of 5 cubic feet per second. Neglecting friction, state how the pressure alters along the axis of the pipe.

18. Take the loss of energy through friction of every pound of water flowing along a straight circular pipe of length l feet and diameter d feet, the velocity being v feet per second, to be $0·0007\ l v^2/d$ foot-pounds. If water at 700 lbs. per square inch conveying 300 horse-power enters a pipe of 6 inches diameter, find the number of cubic feet entering per second and the velocity in the pipe. What is its loss of energy per pound in a length of pipe of 3,000 feet? What is the total loss of power? Obtain an expression for the horse-power lost in transmission in terms of the total horse-power entering the pipe, the entering pressure, p, and l and d.

19. Eight gallons of water per second flow through a 6-inch pipe in which there is a right-angled bend. State the speed. What is the change in the velocity of the water (that is, the *vector* change, as there is no change in mere speed)? What is the change in the momentum of the water per second? What is the resultant force exerted by the water on the pipe at the bend, neglecting friction?

20. A stream is gauged over a rectangular weir or notch, the width of the notch is 3 feet 6 inches, and the height of the still water over the edge of the notch is found to be $14\frac{1}{4}$ inches. Find how many gallons pass over the weir per 24 hours (the velocity of approach may be neglected), and what H.P. could be obtained from turbines supplied by this stream if the available head is 35 feet and the mechanical efficiency of the turbines is 81 per cent. *Ans.* 207,000 gallons and 1,778 H.P.

21. Compare the loss of head by skin friction in a 3-inch pipe with that in a 6-inch pipe, when the velocity of flow is the same in both, and also when the velocity in the small pipe is so increased that it discharges as much water as the big pipe.

22. Water is flowing steadily at a velocity of 3 feet a second through a perfectly horizontal pipe 6 inches in internal diameter; if on a length of $1\frac{1}{2}$ miles there is a drop of pressure owing to friction of 25 lbs. per square inch, what H.P. is being wasted in friction? *Ans.* 3·85 H.P. lost.

23. A large tank, in which a constant head of water is maintained, has a cylindrical hole in the bottom 3 inches in diameter, with sharp edges. How many gallons of water will escape from the tank per hour if the constant head is 9 feet? *Ans.* 564 gals. per hour.

24. Obtain an expression for the loss of head, owing to surface friction, in a long straight pipe through which water is flowing. Hence obtain an expression for the probable discharge in gallons per hour from a main of given diameter and length, when the total head of the water in the reservoir above the point of discharge is known.

25. The quantity of condensing water used by a stationary engine is gauged by first allowing the water to flow into a tank supplied with baffle plates, in order to get rid of any agitation in the water, and then allowing it to flow out of the tank over a rectangular notch 6 inches wide. Estimate the discharge in pounds per minute when the head over the sill is 4 inches. Explain why the flow may be more accurately determined by a V-notch than by a rectangular notch, the head over the notch being supposed subjected to variation.

LECTURE IV.—I.C.E. QUESTIONS.

1. What quantities would it be necessary to measure in order to arrive at the total "head" of each lb. of water flowing in a pipe when it arrives at a given cross-section? Explain how they could be measured. (I.C.E., *Oct.*, 1903.)

2. In a divergent mouthpiece what considerations limit the size at the discharge end relatively to the contracted neck, in order that the mouthpiece may flow full? (I.C.E., *Oct.*, 1903.)

3. Water flows over a rectangular notch 3 feet wide to a depth of 6 inches, and afterwards passes through a triangular right-angled notch; find the depth of water through this notch. The coefficients of discharge for the notches are to be taken as 0·62 and 0·59 respectively. (I.C.E., *Oct.*, 1903.)

4. What is the formula for resistance to the flow of water in a pipe; if the diameter of a pipe is 2 feet 6 inches and the velocity of flow 3 feet per second, find the resistance to flow in a 100-foot length, and the work done per minute in overcoming this resistance. (I.C.E., *Oct.*, 1903.)

5. A semi-circular channel 10 feet in diameter flows full of water; compare its discharge with that of a rectangular channel of the same cross-sectional area 9 feet wide, lined with the same material, and having the same inclination. (I.C.E., *Oct.*, 1903.)

6. Two reservoirs, 10 miles apart, are connected by a pipe 3 feet in diameter, the difference in their water-levels being 40 feet; if the inlet valve to the lower reservoir is partially closed, so that the water rises in a vertical tube, let into the pipe on the inlet side of the valve 20 feet above the level of the water in the reservoir, what would be the discharge of the pipe? (I.C.E., *Oct.*, 1903.)

7. Water impinges on the bucket of a Pelton wheel, with a velocity of 100 feet per second, find the proper peripheral velocity of the wheel to obtain the maximum efficiency, giving proof. (I.C.E., *Oct.*, 1903.)

8. Explain the principle of the hydraulic ram, by means of which a stream of water can be made to pump part of its volume to a higher level. (I.C.E., *Oct.*, 1903.)

9. Water measuring 500,000 gallons per day is diverted from a stream at a point 30 feet above a hydraulic ram which it works; find what quantity of the water the ram can pump to a height of 200 feet above its own level. (I.C.E., *Oct.*, 1903.)

10. Draw a current-meter and describe fully how it can be used to find the discharge of a river. (I.C.E., *Oct.*, 1903.)

11. Write down a formula for the discharge of water over a weir and explain why it takes that particular form. How can the values of the coefficients be ascertained by experiment? (I.C.E., *Feb.*, 1904.)

12. At a drowned orifice 18 inches × 12 inches the water stands 12 inches higher on the approach side than on the downstream side; find the discharge through the orifice, and the width of a rectangular notch necessary to take this quantity of water with a depth of flow of 8 inches, the velocity of approach being negligible. (I.C.E., *Feb.*, 1904.)

13. A pipe 30 inches in diameter branches into two pipes of equal diameter whose combined area equals that of the 30-inch pipe; compare the loss of head in a mile of the latter pipe with that in a mile of the two pipes, the rate of flow being 4 feet per second. (I.C.E., *Feb.*, 1904.)

14. A channel, with a bottom width of 30 feet and side slopes of 2 : 1, flows full of water to a depth of 7 feet; find the velocity of flow in, and discharge of, the channel, the inclination being 1 in 5,000, and c in the Chezy formula 100. (I.C.E., *Feb.*, 1904.)

15. A right-angled elbow, 12 inches diameter, is bolted to the vertical side of a tank containing water, the free surface of which is 20 feet above the centre of the horizontal portion of the elbow. Determine the tension in the connecting bolts, due to the water pressure, when water is being discharged through the elbow under the head in the tank.

(I.C.E., *Oct.*, 1904.)

16. State Bernoulli's theorem. Water flows from an open tank into a vessel in which the pressure remains constant at 2 lbs. per square inch absolute, through a sharp-edged orifice 1 square inch in area. The surface of the water in the tank is kept at 15 feet above the centre of the orifice. Atmospheric pressure 15 lbs. per square inch. Find (i.) the velocity of discharge, (ii.) the discharge in gallons per hour. (I.C.E., *Oct.*, 1904.)

17. Five cubic feet of water are discharged per second from a stationary jet, the sectional area of which is $\frac{1}{2}$ square foot. The water impinges normally on a flat surface, moving in the same direction as the water, with a velocity of 2 feet per second. Find the pressure on the plane in lbs., and the work done on the plane in horse-power. (I.C.E., *Oct.*, 1904.)

18. A canal has a top width of 12 feet, bottom width 6 feet, side slopes 1 to 1, and longitudinal slope 2 feet per mile. Find the discharge per hour. State the value of the coefficient used, or use the value $c = 75$.

(I.C.E., *Oct.*, 1904.)

19. Describe, with the aid of sketches, the method you would adopt to gauge the flow of a small stream, and show how you would determine the quantity of water per hour. State any advantages or disadvantages of the method you adopt. (I.C.E., *Oct.*, 1904.)

20. A district of 10 square miles area drains into a large reservoir. The maximum rate at which the rain falls in the district is $1\frac{1}{2}$ inches in 24 hours. When rain falls after the reservoir is full it is discharged over a weir, the crest of which is level with the high-water level of the reservoir. Find the length of the weir so that the water in the reservoir shall never rise more than 15 inches above its high-water level. (I.C.E., *Oct.*, 1904.)

21. A pipe, 9 inches diameter and 1 mile long, connects two reservoirs. The pipe has a slope of 1 in 80. The level of the water is 25 feet above the inlet end and 6 feet above the outlet end. Neglecting all losses except skin friction, find the discharge and draw the hydraulic gradient. Determine the pressure head in the pipe at a distance of half a mile from the inlet. The coefficient of friction may be taken as 0·007. (I.C.E., *Oct.*, 1904.)

22. How many water mains, 6 inches in diameter, will be required to deliver 1,000 H.P. at a pressure of 700 lbs. per square inch. State carefully the velocity of flow assumed, and the reasons for choosing such a value. Calculate the percentage loss of energy due to friction for each mile length of the main. (I.C.E., *Feb.*, 1905.)

23. A conical pipe varying in diameter from 4 feet 6 inches at the large end to 2 feet at the small end forms part of a horizontal water main. The pressure-head at the large end is found to be 100 feet, and at the small end 96·5 feet. Find the discharge through the pipe. (I.C.E., *Feb.*, 1905.)

24. A pipe consists of half a mile of 12-inch and half a mile of 6-inch pipe, and slopes at 1 in 100. The discharge is 2 cubic feet per second. Find the difference in pressure-head at the two ends of the pipe.

(I.C.E., *Feb.*, 1905.)

25. A cast-iron sewer pipe is 15 inches diameter. It is laid at a slope of 1 in 1,000. How many gallons per day would be discharged—(*a*) when it is flowing full, (*b*) when the depth of the sewage is 6 inches?

(I.C.E., *Feb.*, 1905.)

26. A triangular notch, having an angle of 90 degrees, is used to measure

the flow of a stream. Readings at intervals of 1 hour are taken, as shown in the table:—

Reading.	Head.	Reading.	Head.
1	4 inches.	4	7 inches.
2	5 "	5	6 "
3	6 "		

Draw a curve showing the rate of discharge at any time, and show how you would determine the discharge between the time of the first and last reading. (I.C.E., *Feb.*, 1905.)

27. Water under a head of 60 feet is discharged through a pipe 6 inches diameter and 150 feet long, and then through a nozzle, the area of which is one-tenth the area of the pipe. Neglecting all losses but the friction of the pipe, determine the pressure on a fixed plate placed in front of the nozzle. (I.C.E., *Feb.*, 1905.)

28. A small brass cylinder of uniform diameter, fitted with a non-return valve, is used as a depth-gauge in sounding operations. If after a sounding the top is unscrewed and the cylinder is found to be two-thirds full of water, what is the approximate depth it has reached? (I.C.E., *Oct.*, 1905.)

29. A line of piping has, in the upper portion of its length, a diameter of 15 inches for a length of 5,000 feet and an inclination of 4 per 1,000. A tapering pipe then reduces the diameter to 12 inches, which remains constant for a length of 2,000 feet, throughout which length the inclination is 3 per 1,000. Find the rate of discharge in cubic feet per second when the pipe is fully charged and is delivering freely at its termination. The equation of discharge may be assumed as $q = 42\sqrt{\frac{h d^5}{l}}$, where q denotes cubic feet per second, h the head lost in length l, and d the diameter in feet. (I.C.E., *Oct.*, 1905.)

30. Water is flowing through a circular pipe 4 feet in diameter under atmospheric pressure. At what depth is the discharge a maximum? (I.C.E., *Oct.*, 1905.)

31. A jet of water issues freely from a small orifice in the vertical side of a tank under constant head. The orifice is cut in a thin plate, and there is no loss of head at the orifice. Determine the path of the jet. (I.C.E., *Oct.*, 1905.)

32. Describe in algebraical terms the principle on which the Venturi meter is based. (I.C.E., *Oct.*, 1905.)

33. Describe the principle of action of (i.) a reducing valve, (ii.) an automatic ball air-valve, (iii.) a hydraulic ram for raising water. (I.C.E., *Oct.*, 1905.)

34. A jet of water, having an area of 6 square inches and a velocity of 50 feet per second, strikes perpendicularly a fixed plane of infinite extent. Find the pressure on the plane. Also find the normal pressure on the plane if the axis of the jet is inclined to the plane at an angle of 30°. (I.C.E., *Oct.*, 1905.)

35. A jet of water strikes a curved vane moving in a given direction with a given velocity. Show the form which must be given to the vane at the point at which the water strikes it in order that there may not be any shock. (I.C.E., *Oct.*, 1905.)

36. One hundred horse-power is to be transmitted to a distance of 5 miles with a loss of 15 per cent. of the head due to an accumulator pressure of 750 lbs. per square inch. The beginning and end of the pipe are at the same elevation. Find the diameter of the pipe. The equation of discharge

in Question 29 may be used in this case but with a coefficient of 36 instead of 42. (I.C.E., *Oct.*, 1905.)

37. A pipe A B is fully charged with water at A. Two smaller pipes B C and B D convey the water from B to two points C and D. The length and diameter respectively of A B are 10,000 feet and 15 inches; of B C, 10,000 feet and 12 inches; of C D, 10,000 feet and 9 inches. Points C and D are respectively 50 feet and 80 feet below A. At all points the piping is under pressure except at C and D, where the water issues freely. Find the discharge at C and D, using the equation of discharge in Question 29. (I.C.E., *Oct.*, 1905.)

38. A tank 10 feet square and 10 feet deep has a circular orifice 4 inches in diameter in the bottom, which may be regarded as a thin plate. Water is admitted to the tank until it is full and is then shut off. In how many seconds will the tank be empty? (I.C.E., *Oct.*, 1905.)

39. Give the expression for the total energy of a pound of water flowing in a pipe; if the velocity of the water is 5 feet per second, its pressure 60 lbs. per square inch, and the height of the axis of the pipe at the point in question is 10 feet above the datum, find its value. (I.C.E., *Feb.*, 1906.)

40. A pipe 9 inches diameter gradually contracts to 4 inches diameter, its axis being level. The tops of the two legs of a U-tube containing mercury are connected by tubes to the centres of the two sections, and it is found that the difference in the level of the mercury in the two legs is 6 inches. What quantity of water is flowing through the pipe? The specific gravity of mercury is 13·6. (I.C.E., *Feb.*, 1906.)

41. Find what length of overflow weir is necessary to discharge a rainfall of 1 inch per hour over a watershed of 5,000 acres, if the height of the water over the crest of the weir is to be limited to 18 inches. (I.C.E., *Feb.*, 1906.)

42. Two service reservoirs of equal size, with vertical walls, are connected by a sluice 2 feet square. One of the reservoirs has been emptied for cleaning, and the depth of water in the other one is 20 feet. When the sluice is opened, find how long the water will take to attain the same level in the two if the area of each reservoir is 40,000 square feet. (I.C.E., *Feb.*, 1906.)

43. Derive the formula for the rate of flow of water in a pipe by equating the loss of head to the resistance. Taking a suitable value for the coefficient in the formula for velocity of flow, calculate the loss of head per mile in a pipe 30 inches diameter if the velocity of flow is 5 feet per second. (I.C.E., *Feb.*, 1906.)

44. Water flows out of a reservoir whose surface level is 150 feet above the datum for a distance of 5 miles, through a pipe 3 feet diameter, into a tank whose top water level is 125 feet above datum. Find the velocity of the water in the pipe and the rate of discharge into the tank. (I.C.E., *Feb.*, 1906.)

45. If, in the last question, a valve on the inlet to the lower tank be closed so as to reduce the velocity of flow to one-half of its original value, what would be the reading on a pressure gauge let into the pipe on the approach side of the valve? (I.C.E., *Feb.*, 1906.)

46. Find an expression for the power that can be transmitted by water flowing through a pipe of given size with a constant total fall. What proportion of the head would be lost in friction when the power delivered is a maximum? (I.C.E., *Feb.*, 1906.)

47. Define what is meant by "hydraulic mean depth," and find its value in the case of a channel with a bed-width of 30 feet and side slopes of $1\frac{1}{2}$ horizontal to 1 vertical. If the longitudinal slope is 1 in 6,000 and n in

Kutter's formula equals 0·02, find the rate of discharge of the channel. $c = \frac{\sqrt{m}}{n} \times \frac{x + 1{\cdot}811}{x + \sqrt{m}}$, when $x = n\left(41{\cdot}6 + \frac{0{\cdot}00281}{i}\right)$, m being the hydraulic mean depth and i the inclination. (I.C.E., *Feb.*, 1906.)

48. What is meant by a "forced" vortex? If the wheel of a centrifugal pump is 9 inches inside and 18 inches outside diameter, and revolves at the rate of 1,500 revolutions per minute, to what height is it capable of raising water? Prove the formula made use of. If the pump is provided with a "whirlpool" chamber of twice the diameter of the wheel, to what additional height could the water be lifted? (I.C.E., *Feb.*, 1906.)

49. Define the principal causes of the variation of resistance to the flow of liquids in closed conduits and instance a formula comprising them. Give a general expression for each cause of resistance. Define (*a*) hydrostatic head; (*b*) frictional head; (*c*) hydraulic mean gradient; (*d*) hydraulic mean radius. (I.C.E., *Oct.*, 1906.)

50. Top water-level in a reservoir is 725 feet above datum. Give velocity in feet per second, and discharge in gallons per hour at a point X, 438 feet above datum and 11,976 yards from the reservoir, the main being 6 inches in diameter with a free outlet and an evaluated coefficient of 86. (I.C.E., *Oct.*, 1906.)

51. The discharge calculated in the preceding question decreases by 66 per cent. during the period of minimum flow. Give the pressure at X in lbs. per square inch during that period and the equivalent head in feet, the outlet being regulated to the reduced discharge. (I.C.E., *Oct.*, 1906.)

52. Show graphically the proportional dimensions and radii for egg-shaped sewers. Give the limiting velocities for sewers of all classes. State the rule for determining the size of an egg-shaped sewer from the ascertained diameter of a circular sewer of equivalent capacity. (I.C.E., *Oct.*, 1906.)

53. Give the diameter of a circular brick sewer to run half-full for a population of 80,000, the diurnal volume of sewage being 75 gallons per head, the period of maximum flow 6 hours, and the available fall 1 in 1,000. (I.C.E., *Oct.*, 1906.)

54. State the general formula for the discharge of water over a weir sharp-edged with a free overfall. Find the discharge in gallons per diem for a weir 3 feet wide with a still-water head of 2 inches. (I.C.E., *Oct.*, 1906.)

55. If the discharge last determined was passed through a right-angled triangular notch, what would be the head of water above the vertex of the notch? (I.C.E., *Oct.*, 1906.)

56. Sketch a submerged gauging-weir indicating the up-stream and down-stream heads. State how the discharge is calculated (*a*) by a single formula, and (*b*) by a double formula. (I.C.E., *Oct.*, 1906.)

57. Find the quantity of water discharged in gallons per minute from a circular orifice, 1 inch diameter, under a constant pressure of 34 lbs. per square inch, taking 0·62 as the coefficient of discharge. (I.C.E., *Feb.*, 1907.)

58. (*a*) Define the miner's inch; (*b*) express 10,000 gallons per hour in miner's inches. (I.C.E., *Feb.*, 1907.)

59. The discharge, stated in the previous question, when passed over a sharp-edged rectangular gauging-weir with two end contractions and a free overfall, produces a still-water head of $2\frac{1}{2}$ inches. Find the effective length of the weir. (I.C.E., *Feb.*, 1907.)

60. A rainfall of $\frac{1}{16}$ inch per hour is discharged from a catchment area of 5 square miles. Find the still-water head when this volume flows

over a weir with a free overfall, 30 feet in length, constructed in six bays, each 5 feet wide, taking 0·60 as the coefficient of discharge. (I.C.E., *Feb.*, 1907.)

61. Find the inclination necessary to produce a velocity of $4\frac{1}{2}$ feet per second in a steel water-main, 31 inches diameter, when running full and discharging with a free outlet, using the formula—

$$V = \sqrt[9]{\left(\frac{S \times 2g}{M}\right)^5 \times R^7},$$

in which M, the coefficient of rugosity, has a value of 0·0042. (I.C.E., *Feb.*, 1907.)

62. Calculate the quantity delivered by the water-main in the preceding question per day of 24 hours. This amount, representing the water supply of a city, is discharged into the sewers at the rate of one-half the total daily volume in 6 hours, and is then trebled by rainfall. Find the diameter of a circular brick outfall sewer which will carry off the combined flow when running half-full, the available fall being 1 in 1,500, and taking a value of 0·0054 for M in the formula given above. (I.C.E., *Feb.*, 1907.)

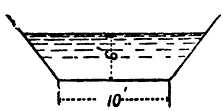
63. Define "momentum" and "moment of momentum." A jet of water issues at a velocity of 100 feet per second from a nozzle 1 inch in diameter. Find the reaction on the nozzle. (I.C.E., *Oct.*, 1907.)

64. A short horizontal pipe connects two vessels. The height of the surface of the water above the centre of the pipe, in one vessel, is maintained at 20 feet, and the vessel is open to the atmosphere. In the second vessel the surface of the water is 3 feet above the centre of the pipe and the pressure on the surface of the water is 3 lbs. per square inch absolute. Find the discharge between the two vessels, taking the coefficient of discharge of the pipe as 0·8. (I.C.E., *Oct.*, 1907.)

65. State a formula for the flow over a sharp-edged weir, explaining the meaning of each term. Describe where and how you would measure accurately the head over the weir, and discuss the effect of (*a*) the form of the approaching channel, (*b*) the velocity of the water in the channel, (*c*) the form of the sill of the weir—on the discharge. (I.C.E., *Oct.*, 1907.)

66. Show that in a channel of uniform section and slope, on the assumption that all the resistance is due to friction on the wetted surface of the channel, the resistance in lbs. per square foot of wetted surface is $F = w \,.\, m \,.\, i$, w being the weight of a cubic foot of water in lbs., m the hydraulic mean depth of the channel, and i the slope of the channel. Hence show, if F is supposed to vary with the square of the mean velocity v, that $v = c\sqrt{m \,.\, i}$. (I.C.E., *Oct.*, 1907.)

67. Find the discharge of a canal of the section shown by the diagram, having a slope of $2\frac{1}{2}$ feet per mile, and side slopes of 1 to 1. The



coefficient c in the formula given in the previous question may be taken as 95. (I.C.E., *Oct.*, 1907.)

68. The rate of flow through a transverse section of a large river is required. Describe carefully with the aid of sketches how you would proceed to determine the flow, and the apparatus you would use. (I.C.E., *Oct.*, 1907.)

75. A jet, issuing from a fire-hose nozzle 1 inch diameter, rises to a height of 121 feet. Find the discharge of the nozzle, assuming the atmospheric resistance is zero. The pressure per square inch in the delivery pump is 100 lbs. per square inch. Determine the fraction of the head in the pump lost between the pump and the nozzle. Find also the force required to hold the nozzle in position. (I.C.E., *Feb.*, 1908.)

ANSWERS TO I.C.E. QUESTIONS.

3. Depth of water through notch = 13·7 inches.

4. $h_L = f\left(\frac{4 l v^2}{d 2 g}\right)$. Resistance to flow, h_L = ·169 foot; and work done per minute in overcoming this resistance = ·282 H.P.

5. Ratio of discharge, 1·065 : 1.

6. Discharge of pipe = 11 cubic feet per second.

7. Peripheral velocity of Pelton wheel = 50 feet per second.

9. Quantity of water pumped by hydraulic ram, allowing an efficiency of 60 per cent., = 45,000 gallons.

12. Discharge from orifice = 7·44 cubic feet per second; width of rectangular notch = 4·13 feet.

13. The loss of head in each case is 15·84 feet and 44·8 feet.

14. The velocity of flow of the channel = 3·17 feet per second; discharge of channel = 750·4 cubic feet per second.

15. Tension in the connecting bolts = 1,960 lbs.

16. (i.) Velocity of discharge = 53·8 × ·97 = 52·1 feet per second; (ii.) discharge = 5,200 gallons per hour.

17. Pressure on the plane = 77·5 lbs.; work done per minute on the plane = ·28 H.P.

18. Discharge = 193,700 cubic feet per hour.

20. Length of weir = 87·3 feet, when using a coefficient of discharge of ·62.

21. Discharge of water from pipe = 2·3 cubic feet per second; hydraulic gradient is 1 in 62·4; pressure head in the pipe at a distance of half a mile from the inlet = 15·3 feet.

22. Assumed velocity of flow = 3 feet per second; number of water mains required = 10; percentage loss of energy for each mile length of the main due to friction = 2·75 per cent.

23. Velocity of flow = 15·32 feet per second; discharge through the pipe = 48·16 cubic feet per second.

24. Difference in pressure head at the two ends of the pipe = 211·8 feet.

25. Discharge (*a*) 1,067,000 gallons per day; (*b*) 388,000 gallons per day.

27. Pressure on fixed plate placed in front of nozzle = 135 lbs.

29. Rate of discharge from pipe = 3·55 cubic feet per second.

30. Depth of maximum discharge of water from a circular pipe = 3·8 feet.

31. The path of the jet is a parabola whose equation is $y = \frac{g x^2}{4 g h}$, where h is the head.

34. Pressure on the plane = 203 lbs.; normal pressure on the plane, when the axis of the jet is inclined at an angle of 30° to the plane, = 101·7 lbs.

36. Diameter of pipe = 5·88 inches.

37. Discharge at C and D = 1·82 and 1·43 cubic feet per second respectively.

38. Time taken to empty the tank = 24·35 minutes, when using a coefficient of discharge of ·62.

HELPS AND SOLUTIONS TO INST.C.E. QUESTIONS.

LECTURE IV.—A.M.INST.C.E. QUESTIONS.

1. Deduce a general expression for the velocity of flow in an open channel. An irrigation channel, with side slopes at 1 to 1, has a bottom width of 100 feet and a depth of 10 feet. It discharges 3,000 cubic feet per second. Find the slope in feet per mile.

Let P_1 = the pressure per square foot of cross-section at any part of the channel.
,, P_2 = the pressure per square foot at a point l feet lower down stream.
,, A = cross-sectional area of the channel in square feet.
,, S = wetted surface of the channel.
,, f = a coefficient depending on the roughness of the channel.
,, v = velocity of flow in feet per second.

Now, by the laws of fluid friction, the frictional resistance $= fSv^2$, and the pressure producing flow $= (P_1 - P_2)A$.

Hence, for steady flow, $(P_1 - P_2)A = fSv^2$,

or,
$$\frac{P_1 - P_2}{w} = \frac{f}{w} \times \frac{sl}{A} \times v^2,$$

where s is the wetted perimeter, or

$$\text{the pressure head lost in friction} = \frac{f}{w} \times \frac{1}{m} \times v^2 l,$$

where $m = \frac{A}{s}$, the hydraulic mean depth.

If, therefore, h is the head lost in friction,

$$v^2 = \frac{w}{f} \times m \times \frac{h}{l},$$

or,
$$v = c\sqrt{m\,i},$$

where $c = \frac{w}{f}$ is determined experimentally for different channels, and i is the hydraulic gradient.

In the example, $A = \left(\frac{120 + 100}{2}\right) 10 = 1{,}100$ sq. ft.

And, $s = 2\sqrt{200} + 100 \doteqdot 128{\cdot}3$ feet;

$\therefore$ hydraulic mean depth, $m = \frac{A}{s} = \frac{1{,}100}{128{\cdot}3} = 8{\cdot}57.$

Also, $v = \frac{3{,}000}{1{,}100} = \frac{30}{11}$ feet per second.

Assuming $c = 130$, we get $\quad v = c\sqrt{m\,i}$,

$$\frac{30}{11} = 130\sqrt{8{\cdot}57\,i},$$

$$\therefore \qquad i = \left(\frac{30}{11 \times 130}\right)^2 \times \frac{1}{8{\cdot}57} = 0{\cdot}00005135.$$

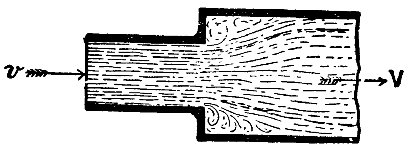
That is, $\quad i = \frac{h}{l} = 0{\cdot}00005135.$

When $l = 5{,}280$ feet, then $\quad h = l \times 0{\cdot}00005135$

$$h = 5{,}280 \times 0{\cdot}00005135 = 0{\cdot}27 \text{ foot},$$

whence, $\quad$ the slope, $i = \frac{h}{l} =$ **0·27 foot per mile.**

2. Give an expression for the loss of head in a pipe owing to a sudden enlargement of section.



Loss of Head due to Sudden Enlargement in a Pipe.

Let v = velocity in small part of the pipe,
„ V = velocity in large part of the pipe,
„ a = cross-sectional area of the small part of the pipe,
„ A = „ „ large „

Then we have first $av = AV$.

If the pipe were enlarged very gradually the water would change its velocity quietly without any eddying motions being set up in it. Actually, what happens is this, that on opening into the large pipe the water breaks away in all directions; there is a sort of main stream flowing on, but the corners are full of broken water, which is continually joining the main stream, and its place being supplied from the small stream entering. The water flowing across the larger section at entry has all kinds of cross motions, and head is wasted in producing these motions.

The effect is due to a stream moving with a velocity v impinging on a larger stream moving at velocity V, the relative velocity or velocity of striking being $v - V$.

Hence, the loss of energy per pound of water, or the waste of head due to abrupt change of section $= \dfrac{(v - V)^2}{2g}$.

If $A = na$, the water in the large section is moving $\frac{1}{n}$th as fast as in the small section. Then the loss of energy per pound of water, or the loss of head when a pipe suddenly enlarges n times is $\dfrac{v\left(1 - \frac{1}{n}\right)^2}{2g}$. Or, if we refer

to the velocity in the large section as V, we have the velocity in the small section n V, and

$$\text{the loss of head will be} = \frac{(n\,V - V)^2}{2g} = \frac{V^2}{2g}(n-1)^2.$$

3. Obtain an expression for the loss of head due to a sudden enlargement in a pipe, and calculate the loss in the case of a pipe 2 inches in internal diameter suddenly enlarged to 4 inches, the discharge being 5,000 gallons per hour.

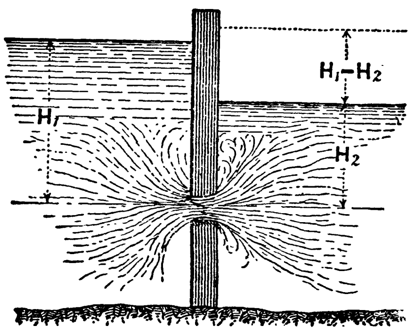
Here, $$V = \frac{Q}{A} = \frac{5{,}000 \times 28 \times 144}{3{,}600 \times 22 \times 16 \times 6{\cdot}25} = 2{\cdot}546 \text{ feet per second.}$$

and $$n = \frac{A}{a} = \left(\frac{D}{d}\right)^2 = 4.$$

$$\therefore \left.\begin{array}{r}\text{Loss of head due}\\ \text{to enlargement}\end{array}\right\} = \frac{V^2}{2g}(n-1)^2 = \frac{(2{\cdot}546)^2}{64}(4-1)^2 = \frac{(2{\cdot}546)^2 \times 9}{64}$$

$$\text{,,} \quad \text{,,} \quad = 0{\cdot}9116 \text{ foot} \doteqdot \mathbf{10{\cdot}94 \text{ inches.}}$$

4. Give a formula for the discharge of drowned orifices. What advantage do they offer for gauging purposes ?



Drowned Orifice.

Let H_1 = head of water on one side of orifice.
,, H_2 = ,, on other side of orifice.
,, A = cross-sectional area of the orifice.

Then, $$H_1 - H_2 = \text{head producing flow,}$$

and $$\text{velocity of flow, } V = \sqrt{2g(H_1 - H_2)}.$$

$$\therefore \quad \text{Discharge of drowned orifice, } Q = K\,A\,\sqrt{2g(H_1 - H_2)},$$

where K is a coefficient discharge which varies, but 0·62 is taken as the mean value.

The advantage which drowned orifices offer for gauging purposes is, that the theoretical discharge from a submerged orifice is the same for the same effective head, whatever be its distance below the lower water level.

It is not likely that the same coefficients of discharge would be found for deeply submerged orifices as for those slightly submerged, but experiments in this direction are lacking from which any definite conclusions could be drawn.

5. A pipe tapers to one-tenth its original area, and then widens out again to its former size. Calculate the reduction of pressure, at the neck, of the water flowing through it in terms of the area of the pipe and the velocity of the water. Why is this reduction of pressure a gauge of the discharge?

The principle of the Venturi meter is worthy of attention. From the observed reduction of pressure caused by a contraction in a pipe, the velocity, and thence the rate at which the water is delivered through it, is inferred as a question of hydrodynamics.

Let v_1 be the velocity, and p_1 the pressure in the pipe, and p_2 the pressure in the contracted portion; and if A is the area of the pipe, a that of the contracted portions, and w the weight of unit volume of water,

$$\frac{v_1^2}{2g} + \frac{p_1}{w} = \frac{\left(\frac{v_1 A}{a}\right)^2}{2g} + \frac{p_2}{w}.$$

That is, $$p_1 - p_2 = \frac{w\,v_1^2}{2g}\left\{\left(\frac{A}{a}\right)^2 - 1\right\}.$$

Also, the discharge, $$Q = A\,v_1 = \frac{A\,a}{\sqrt{A^2 - a^2}}\sqrt{2g\frac{(p_1 - p_2)}{w}} = \frac{A\,a}{\sqrt{A^2 - a^2}}\sqrt{2g\,H},$$

where H denotes the difference of pressure head at the throat and up-stream large end of the cone.

Due to friction, &c., the discharge,

$$Q = k\,.\,\frac{A\,a}{\sqrt{A^2 - a^2}}\sqrt{2g\,H},$$

where k is a coefficient determined by experiment.

From this it follows that the velocity in the pipe is proportional to the square root of the reduction of pressure.

In the example, $A = 10\,a$,

$$p_1 - p_2 = \frac{62 \cdot 5\,v_1^2}{64}\{100 - 1\} = 96 \cdot 68\,v_1^2 \text{ lbs. per sq. ft.,}$$

$$\text{or } 0 \cdot 67\,v_1^2 \text{ lbs. per sq. inch.}$$

6. A rectangular chamber 120 feet square contains 15 feet depth of water, which is allowed to flow out through a vertical rectangular orifice 2 feet by 1 foot, the top of which is level with the floor of the reservoir and the tail-water. Calculate the time it will take to empty.

Let y be the head after t seconds from the commencement of flow, and let the water descend δy feet in the time δt.

The discharge in time δt is $= 120 \times 120 \times \delta y$ cubic feet.

But the discharge is also $= 0 \cdot 62 \times 2$ (sq. ft.) $\times \sqrt{2gy} \times \delta t$,

where $0 \cdot 62$ is taken as the coefficient of discharge.

$$\therefore \quad 120 \times 120 \times \delta y = 0{\cdot}62 \times 2 \times 8\sqrt{y} \times \delta t = 9{\cdot}92\sqrt{y} \,.\, \delta t$$

Or,
$$\delta t = \frac{14{,}400}{9{\cdot}92} y^{-\frac{1}{2}} \delta y = \frac{450}{0{\cdot}31} y^{-\frac{1}{2}} \,.\, \delta y.$$

Or,
$$dt = \frac{450}{0{\cdot}31} \int_0^{15} y^{-\frac{1}{2}} dy.$$

$$\therefore \quad t = \frac{900}{0{\cdot}31} \left[y^{\frac{1}{2}} \right]_0^{15} = \frac{900}{0{\cdot}31} \sqrt{15} = 11{,}244 \text{ seconds.}$$

$$t = \textbf{3 hours 7 minutes.}$$

7. Water issues from the nozzle of a fire-engine, $1\frac{1}{2}$ inches diameter, in a jet which rises to a height of 100 feet. Neglecting the contraction of the jet, find the reaction on the machine due to the jet.

When a stream or jet is in motion delivering W pounds of water per second, with the uniform velocity v, that motion may be regarded as produced by a constant impulsive force F, which has acted upon W for one second and then ceased. In this second the velocity of W has increased from o to v, and the distance $\frac{1}{2}v$ has been described. Consequently the work $F \times \frac{1}{2}v$ has been imparted to the water by the impulse F. But the theoretic energy of the jet is $W\frac{v^2}{2g}$; hence

$$F \times \tfrac{1}{2}v = W\frac{v^2}{2g},$$

or, force of impulse, $F = W\dfrac{v}{g}$.

Let a be the area of cross-section of the jet; then $W = wav$,

and
$$F = wa\frac{v^2}{g}.$$

The reaction of a jet upon a vessel occurs when water flows from an orifice. This reaction must be equal in value and opposite in direction to the impulse. In the direction of the jet the impulse produces motion, in the opposite direction it produces a pressure which tends to move the vessel. Hence the force of reaction of a jet is

$$F = W\frac{v}{g} = wa\frac{v^2}{g}.$$

To compare this with hydrostatic pressure, let h be the velocity-head due to v; then
$$F = 2wa\frac{v^2}{2g} = 2wah.$$

But the normal pressure on a surface of area a under the hydrostatic head h is wah. Therefore, the dynamic pressure caused by the reaction of a jet issuing from an orifice in a vessel is double the hydrostatic pressure on the orifice when closed. This theoretic conclusion has been verified by experiment.

In the example, $w = 62 \cdot 5$ lbs. per cubic foot; $a = \frac{22}{28} \times \frac{9}{4 \times 144}$ square, feet; $h = 100$ feet.

$\therefore$ Reaction on machine due to jet, $F = 2\,w\,a\,h = 2 \times 62 \cdot 5 \times \frac{11 \times 100}{28 \times 32}$

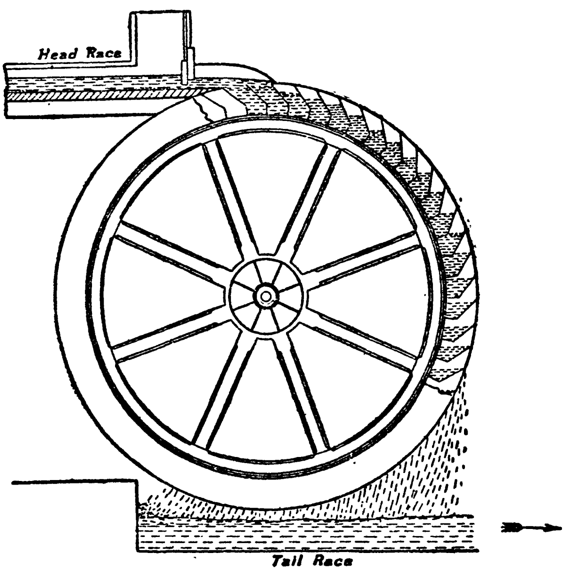
" " " $F = 153 \cdot 5$ lbs.

LECTURE V.

WATER-WHEELS AND TURBINES.

CONTENTS.—Hydraulic Motors—Overshot Water-wheel—Breast-wheels—Undershot Water-wheel—Fairbairn's Improvements—Clack Mill—Pelton Wheel—Turbines—Girard Turbine—Jonval Turbine—Günther's Governor—Thomson's Vortex Turbine—Little Giant Turbine—Hercules Mixed-Flow Turbine—Centrifugal Pumps and Fans—Efficiency of Pumps and Turbines—Questions.

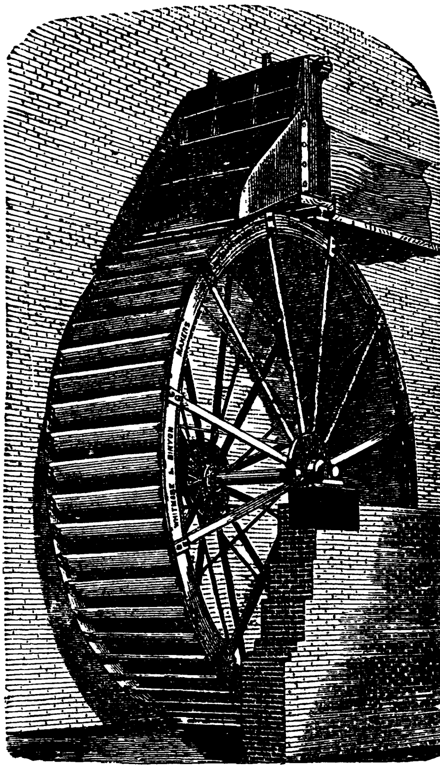
Hydraulic Motors.—In connection with hydrostatics we have



OVERSHOT WATER-WHEEL.

already described some machines for obtaining motion by hydraulic means, but in all these cases the water acts solely by its pressure,

We now come tó the consideration of other water motors in which the weight and momentum of the water are also employed. These may be divided roughly into two classes—Water-wheels, in which the water acts for the most part directly by its weight; and turbines, in which it acts by its momentum. We cannot, however, draw a very sharp line between them, as they gradually merge into one another, and the power of both ultimately depends on gravity.

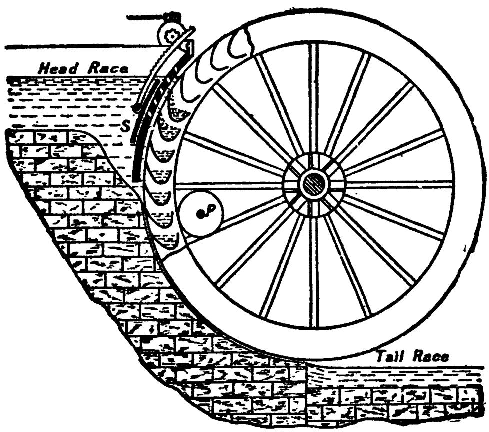


OVERSHOT WATER-WHEEL BY MESSRS. WHITMORE & BINYON.

Overshot Water-wheel.—This consists of a wooden or iron frame to which is fixed a number of blades, so as to form with the inner circumference a series of buckets for holding water. The water is led along an aqueduct termed the *head race* to the top of the wheel, and there enters the buckets. Its weight forces them downwards and thus makes the wheel revolve. As each bucket in turn approaches its lowest position the water gradually drops out of it into the tail race. The motion of the water as it enters the wheel also assists to some extent in producing rotation. The buckets are so shaped and fixed to the wheel, that as little as possible of the available power shall be lost by the water spilling from them before they reach the tail race. The sectional view shows one form of bucket for this purpose whilst the outside view shows another with curved blades. One disadvantage of this wheel is, that the water leaves

it with a velocity opposite in direction to that of the tail race, if this flows the same way as the head race. The water therefore does not get away so freely from the tail race, and more clearance is necessary at the bottom of the wheel which thereby involves a loss of head.

Breast-wheels.—This last consideration, coupled with the difficulty of supporting the head race for large overshot wheels, has brought about the introduction of breast-wheels, in which the water is introduced between the top and middle of the wheel as shown by the next illustration. The breast-wheel is also frequently made with curved blades into which the water drops almost vertically, and then acts chiefly by its weight. The motion of the

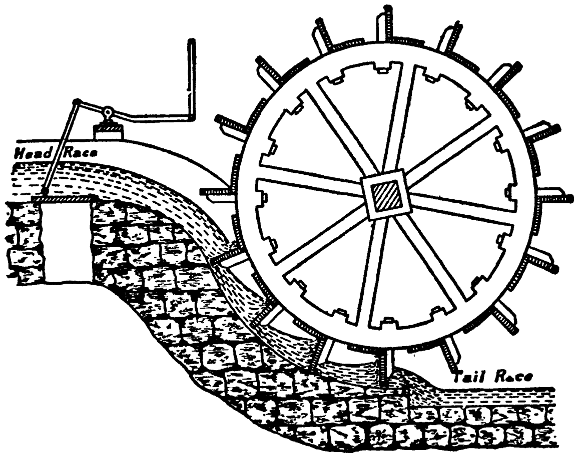


FAIRBAIRN'S BREAST-WHEEL.

wheel in this case assists the escape of the tail water instead of hindering it as in the previous one. The curved ventilated form of bucket with closed breast, as shown in the above figure, was first introduced by Sir William Fairbairn and greatly increased the efficiency of the motor. But, for small wheels, such as are used for farms, where first cost is more important than efficiency, they are usually made radial as shown by the following figure.

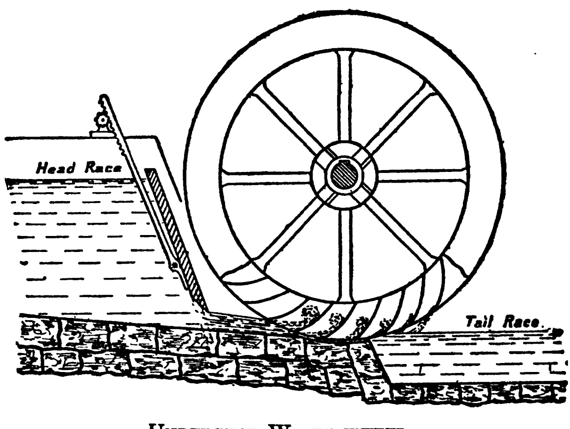
Here, the water is allowed to attain a certain amount of motion before reaching the wheel, and therefore acts partly by momentum and partly by its weight. The buckets have no ends, but the wooden breast serves to keep the water from escaping by the sides and circumference of the wheel before it reaches the bottom.

Breast-wheels into which the water enters near the top are called high breast-wheels.



BREAST-WHEEL WITH RADIAL BLADES AS USED IN COUNTRY FARMS.

Undershot Water-wheel.—The breast-wheel just described forms a connecting link between water-wheels proper and turbines, to

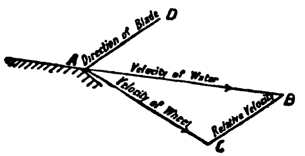


UNDERSHOT WATER-WHEEL.

which latter class undershot wheels really belong. With an undershot wheel the whole energy of the water is allowed to become kinetic and then it acts on the blades solely by its momentum. Here,

again, radial blades are common, but they are very inefficient on account of the large loss due to the eddies which are caused by the impact of the water upon them, and from the kinetic energy which the water still possesses on leaving the wheel.

In order that the water may be applied without shock and to avoid the formation of eddies, the tips of the blades should be so inclined as to be parallel to the motion of the water relatively to the wheel at its point of entering. To find this direction, draw A B to represent the velocity of the water as it leaves the head race, and A C the circumferential velocity of the wheel at the point where the water enters it. Then, by completing the triangle A B C we find C B the direction of the tip of the blade at A. The blade is curved upwards, and the position where the water leaves it is at the same level as where it enters, in order that the water may drop out with as little kinetic energy as possible.



ANGLE OF BLADE.

Fairbairn's Improvements.—In the illustration of Fairbairn's breast-wheel, we have shown the regulating sluice S connected by a curved rack and pinion to the worm gear which is turned by a hand-wheel, until the sluice admits the desired quantity of water to develop the necessary power. In the case of large wheels which have to drive textile or other machinery requiring great uniformity of speed, this worm gear is connected to a ball governor which automatically adjusts the position of the regulating sluice, to suit the different demands of the works.

In addition to other improvements effected by Fairbairn, we may mention, that instead of driving from a spur-wheel keyed to the water-wheel shaft, he bolted a segmental annular toothed wheel directly to one of its outer sides or flanges and geared it with a pinion as shown at P. He thus diminished the distance between the plummer block bearings of the water-wheel shaft, relieved its radial arms from conveying the driving stresses, and at once obtained the necessary speed without intermediate gearing. The importance of this improvement was so self-evident to mill-owners, that many large wheels which had previously given considerable trouble and shown signs of distress, were fitted with Fairbairn's drive, and are working to the present day at full power with perfect regularity and freedom from breakdowns.

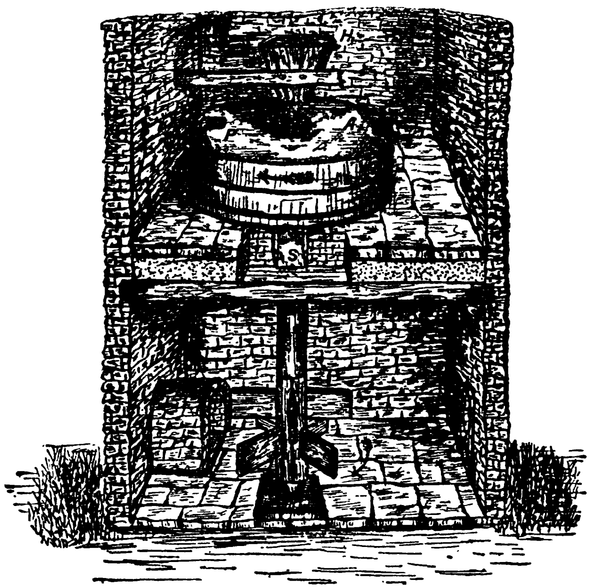
We may here summarise Fairbairn's improvements in water-wheels, viz. :—(1) The use of iron instead of wood in their construction, thus lightening and at the same time strengthening the various parts.

(2) The adoption of curved ventilated iron buckets instead of straight wooden ones, to prevent eddies and thus obtain a greater efficiency from the head and body of water.

(3) The introduction of a closed breast to prevent the escape of the water during its turning of the wheel.

(4) Driving directly from the sides and periphery of the water-wheel in order to minimise the stresses in the arms, reduce the distance between the bearings, and at once obtain the desired speed.

The Clack Mill.—One of the oldest forms of water-wheels, and one which belongs to the same class as the undershot wheel, is

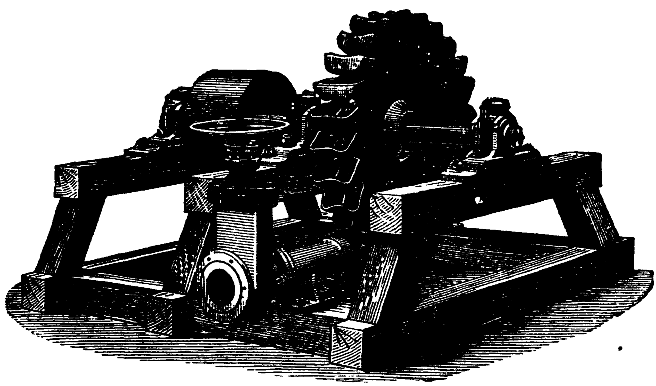


THE OLD CLACK MILL.

that known as the "clack mill." It is supposed to be of Norwegian origin, and was certainly introduced therefrom into the Orkney and Shetland Islands several hundred years ago. Our illustration is reduced from a sketch made by Mr. Sellar in 1898 of the only remaining working clack mill in Orkney, at the farm of Millbrig, in the parish of Birsay. The owner, Mr. Folster, says that it has been handed down to him from generation to generation for more than two hundred years.

The water under a head of 10 feet enters by the left-hand aperture and leaves by the right-hand one. During its passage the water impinges upon the further side of the wooden radial arms which are fixed to the vertical iron shaft. The lower end of this shaft rests in a conical footstep, whilst the upper end carries the revolving millstone, and is steadied by a bearing immediately beneath the lower or fixed stone. The corn to be ground is fed into a "head" or "harp" H, and as the upper stone revolves the projecting wooden pin P, strikes the radial arm A connected to the lower end of H. This shoggles a portion of the grain into the central opening at each revolution. In doing so, a clacking noise is made by this pin striking the outstanding arm, which has given rise to the local term of "clack mill." The grain is carried down by gravity and finds its way into groves between the two stones where it is ground into rough meal. This compound of meal and husk dribbles from the shoot S into a wooden bin, from which it is removed, and separated by shaking and blowing, or by another machine. for future use in the shape of porridge or oat-cake.

Pelton Wheel.—The previous examples naturally lead us to the consideration of the Pelton wheel, which is very often used for



PELTON WATER-WHEEL BY MESSRS. W. GÜNTHER & SONS, OLDHAM.

great pressures and high falls. As will readily be understood from a consideration of the accompanying figure, this form of water-wheel consists of a plain disc mounted upon a central shaft, and carrying a number of curved buckets fixed at equal distances around its periphery. A conical nozzle attached to the supply pipe is so fixed as to direct a jet of water upon each of these buckets in turn and thus drive the wheel at a high speed. The

buckets have a central division, and as they curve outward towards each side, the jet is thereby deflected in two portions and then backwards. With properly designed buckets, and when the circumferential velocity of the wheel is half that of the jet, the water will simply leave the buckets with little or no remaining kinetic energy, and hence the efficiency of such a wheel may be very great. Pelton wheels are used for falls having a head of from 30 to 2,000 feet, or for corresponding pressures derived from water-pumps and city hydraulic power mains. They are sometimes made with several nozzles, each being fitted with a stop-valve, so that the power can be varied by shutting off the water from one or more of them. When there is only one jet, the power can be varied without a change of efficiency by simply unscrewing the nozzle and putting on another of a different size. If the power is varied by partially closing the stop valve, we lose a large amount of energy in friction at the valve, and the efficiency is thereby considerably reduced.

We may find the efficiency of a Pelton wheel, on the assumption that there are no eddies or friction, as follows :—

Let V = The velocity of the water as it issues from the nozzle.
„ v = „ „ vane.

Then, $V - v$ is the relative velocity of water on vane, which is not changed as it moves around the vane. Therefore, the final velocity of the water is $v - (V - v)$ or $2v - V$, which may be either positive or negative. Now, by the principle of conservation of energy, the initial energy of the water is equal to that spent on the vane + the final energy. Therefore, the efficiency of the wheel is :—

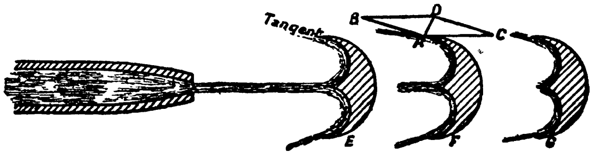
$$\eta = \frac{\textit{Power got out}}{\textit{Power put in}} = \frac{\textit{Initial kinetic energy} - \textit{final kinetic energy}}{\textit{Initial kinetic energy}}$$

$$= \frac{V^2 - (2v - V)^2}{V^2} = \frac{4v(V - v)}{V^2}.$$

This is evidently greatest, and equal to unity, when $2v - V$ is zero, or $V = 2v$, since $(2v - V)^2$ can never be negative. That is, for maximum efficiency the speed of the wheel should be half that of the water, and when it runs either quicker or slower than this the efficiency will be less. For example, if the speed of the jet is 100 feet per second and the vanes travel 20 feet per second, the theoretical maximum efficiency is :—

$$\eta = \frac{4 \times 20(100 - 20)}{100 \times 100} = 0{\cdot}64 \text{ or 64 per cent.}$$

The above are ideal cases in which the water is directed backward by the vane parallel to its original direction. In practice, the Pelton vanes do not quite do this, but are slightly divergent, as shown, in order that the water may clear the following vane.



VANES FOR PELTON WHEEL.

The final velocity of the water is easily found by the parallelogram of velocities, thus—Draw A B the velocity of the water and A C that of the vane. Then, A D is the final velocity of the water, and the kinetic energy carried away is proportional to A D^2.

We have seen, in the preceding Lecture, that when the ideal vane is stationary, the pressure on it is four times the statical pressure on the area of the jet. At maximum efficiency, when the relative velocity of the water and vane is half what it is when the vane is stationary, the force on the vane is equal to this statical pressure, because the impact varies as the square of the relative velocity. Or,

$$\text{Water striking vane per second} = \frac{w}{g} a (V - v) \text{ lbs.}$$

$$\text{Change of velocity produced by vane} = 2 (V - v).$$

$$\therefore \quad \text{Momentum imparted per second} = \frac{w}{g} a (V - v) \times 2 (V - v).$$

$$\text{Or, Force of jet on vane} \quad . \quad . \quad . \quad . \quad = \frac{2 w a}{g} (V - v)^2.$$

$$\text{And, when,} \qquad v = \tfrac{1}{2} V = \tfrac{1}{2} \sqrt{2 g h},$$

$$\text{Then,} \qquad F = \frac{2 w a}{g} (\tfrac{1}{2} \sqrt{2 g h})^2 = w a h.$$

Hence, in this case, the power expended on the vane is $F \times \frac{1}{2} v$. But, the total power of the jet is $F \times v$, or twice as much. This would make the efficiency to be one half, but we have proved it to be unity. Where is the discrepancy?

Suppose a single vane to move away indefinitely in a straight line in the direction of the jet, it will be seen that there is an ever-increasing quantity of water flying through the air after the

vane. This represents an ever-increasing debt of uncollected kinetic energy, so that only half the water issuing per second strikes the vane in the same time.

But now, let E be the initial position of the vane (see previous figure), and E F the distance it travels in one second. When it has arrived at F, let another vane be suddenly interposed at E. There is now a rod of water between E and F running twice as fast as the vanes, and the last particle of this water will only overtake F when it arrives at G and the second vane at F. A third vane is now interposed at E and the process repeated.

This is what happens in the Pelton and other impact wheels, such as the Laval steam turbine. The fluid strikes two vanes at once, one behind the other; a statement hard to believe unless approached by the above argument.

Turbines.—These form a type of water motor which occupy much less space, are more efficient, more easily governed, suit a greater range of fall, and generally run at a greater speed than ordinary water-wheels. They are classified in several different ways according to the manner in which their special properties are considered. We shall first of all divide them into four classes, viz.—(1) inward flow; (2) outward flow; (3) parallel (or axial) flow; and (4) mixed flow turbines. In the first two kinds, the water either flows inwards from the outer circumference as in the Thomson turbine, or outwards from the inner circumference as in one type of the Girard. In the third class, it flows parallel to the axis, entering at one side of the wheel and leaving at the other as in the Jonval. The fourth type has both radial and axial flow, and the water usually enters at the circumference and leaves at one or both sides parallel to the axis.

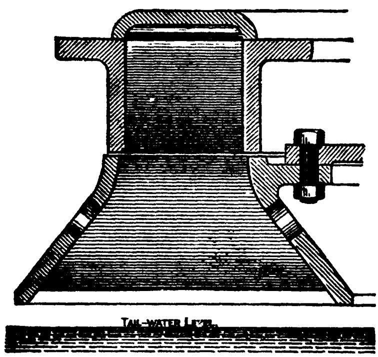
In the second place, we find all of the previous types divided into what are termed *drowned* and *ventilated* turbines. The former are designed to run quite full of water and may be submerged in the tail race, or used with a suction pipe; whilst the latter have ducts to admit air at atmospheric pressure into the wheel, and, consequently, cannot be submerged or used with such a pipe. The energy of the water entering the drowned type is partially potential and partially kinetic, whilst it is wholly kinetic before it enters a ventilated turbine. Hence, these two kinds of motors are sometimes respectively called *reaction* and *impulse* turbines.

The purpose of a suction pipe is to allow the turbine to be placed some distance above the tail race without losing the corresponding head. This suction pipe is merely an air-tight conduit to carry away the used water and must have its lower end below the surface of the tail race; moreover, it must not exceed 20 feet or thereby in vertical height from the turbine to the tail race; otherwise

the water column supported by the pressure of the atmosphere will be broken by the introduction of air into the exhaust conduit.

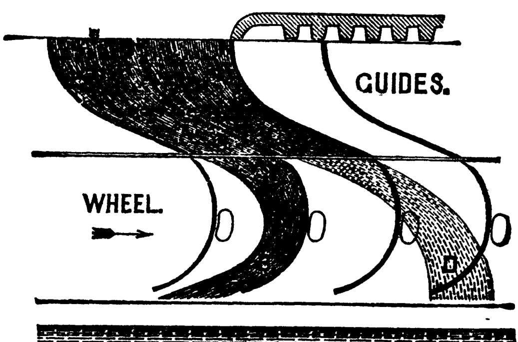
A disadvantage of most turbines of the "drowned" type is, that in regulating its speed and power we cannot gradually cut off the water without a considerable drop in efficiency. We can, however, do so in several steps when divisions are placed in the wheel for this purpose. This is because the guide passages to the wheel must always be quite full of water, which would not be the case if the opening to any particular guide port was only partially closed.

Girard Turbine.—This wheel is named after the French engineer M. Girard, who in 1856 first designed the ventilated and impulse turbine. It can be made with either a vertical or horizontal axis and for both axial and outward flow. The former is used for low falls of from 6 feet and upwards, and the latter for high falls up to 1,000 feet.

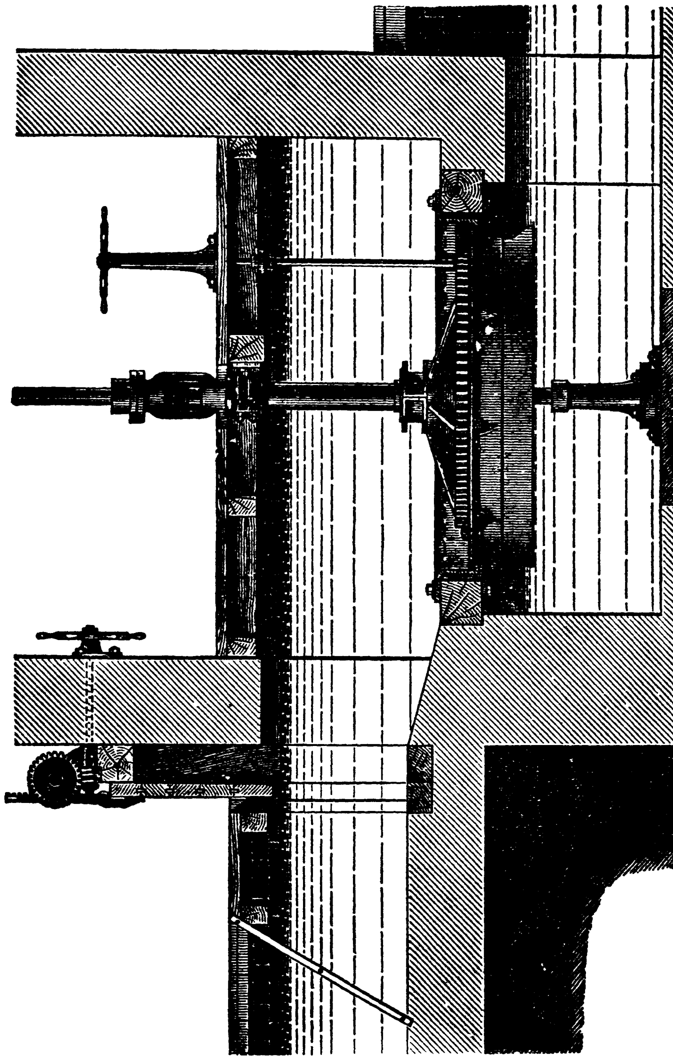


RADIAL SECTION. GÜNTHER'S AXIAL FLOW GIRARD TURBINE.

By referring to the two accompanying figures, which represent radial and circumferential sections of this turbine, it will be seen that the gate for controlling the water supply is placed above the guide ports A, C, through which the water passes and issues with full velocity due to its

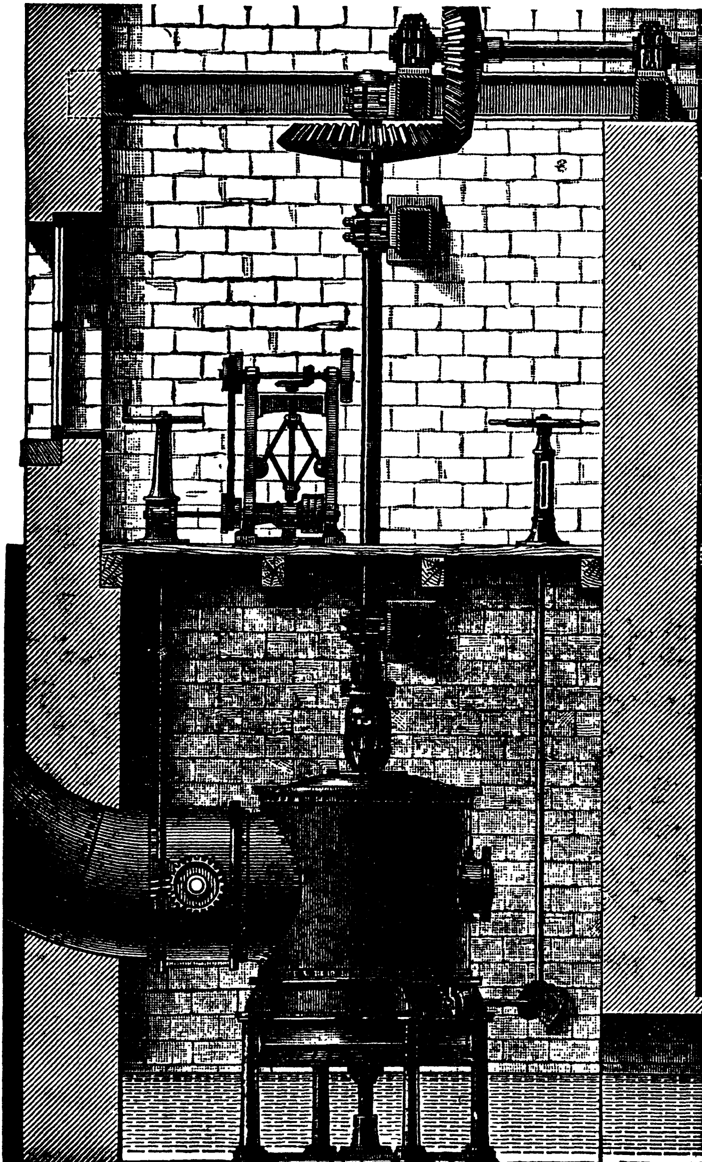


CIRCUMFERENTIAL SECTION. GÜNTHER'S AXIAL FLOW GIRARD TURBINE.



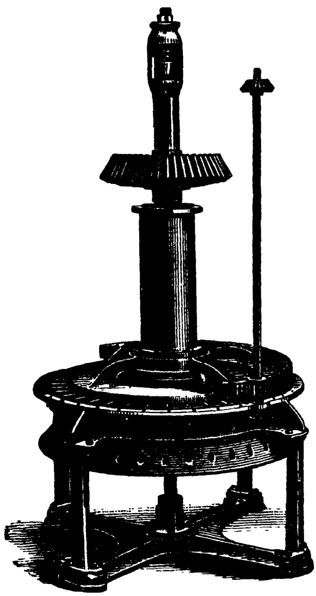
Girard Turbine for Low Falls by Messrs. W. Günther & Sons, Oldham.

head. It then glides along the concave surfaces of the wheel buckets B, but does not quite fill them. The inclination of the upper edges of the buckets is obtained in the manner explained

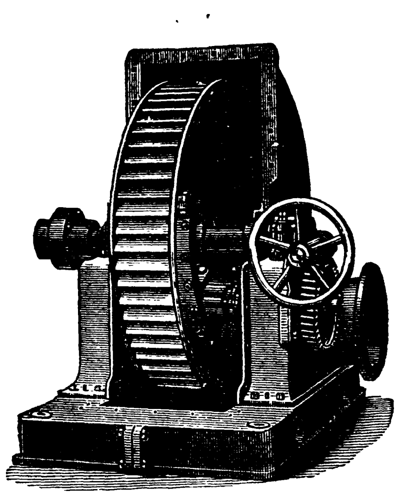


GÜNTHER'S GIRARD TURBINE FOR LOW FALLS.

for undershot wheels, while that of the lower edges is made as small as possible in order that the water may not leave the vanes with much kinetic energy. To allow of this inclination being smaller than would otherwise be the case, the sides of the wheel are splayed outwards as shown by the radial section in order that the water may spread and not foul the convex surface of the next blade. The path of the water relatively to the moving wheel is shown at B, whereas, the dotted lines through D show the actual motion of the stream as seen from a fixed point. The ventilating holes are clearly shown in both views. These



GÜNTHER'S GIRARD TURBINE FOR LOW FALLS.

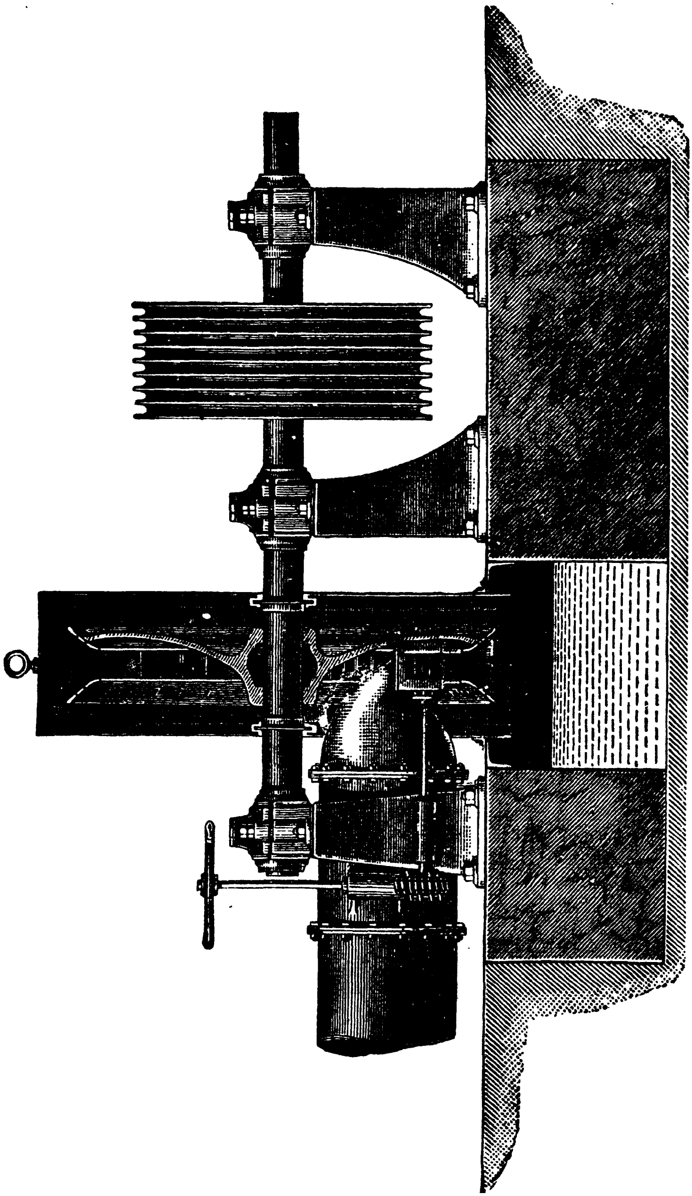


GÜNTHER'S GIRARD TURBINE FOR HIGH FALLS.

admit air to the wheel and prevent the formation of eddies in the empty parts.

The above left-hand illustration is that of a complete turbine, and the two full-page illustrations show how they are fixed in position ready for work. The second of these also indicates how the governor is attached.

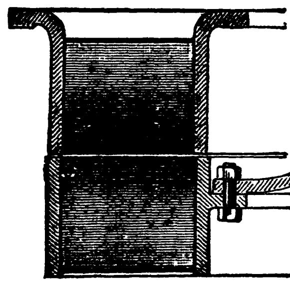
When used for high falls the water is only admitted to a few of the buckets at a time. It is then usually made with outward



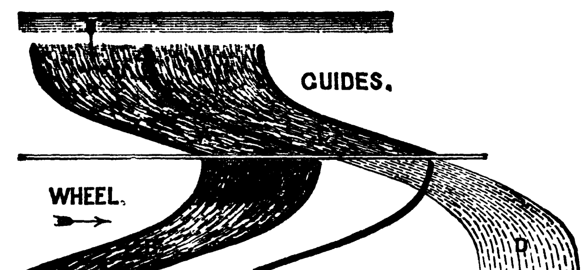
Longitudinal Section of Günther's Girard Turbine for High Falls.

flow, and mounted on a horizontal axis as illustrated by the sectional drawing facing this and the right-hand figure on the previous page. The pipe for admitting the water and the hand gear for adjusting the flow are clearly visible, whilst a grooved pulley for rope driving is placed on the right. This type of turbine gives a high efficiency at medium as well as at full load.

Jonval Turbine.—This turbine is of the parallel or axial flow type and is designed to be always kept full of water. The accompanying figures give radial and circumferential sections, having the same lettering as the corresponding figures for the Girard turbine. It is usually employed for low and medium falls of from 2 to 40 feet, and can run equally well when completely submerged or when connected to a suction pipe. The adjustment of the water supply is effected by means of a slide or slides which close the guide passages one after another. A turbine of this type has the greatest efficiency when all its passages are full of water, and consequently the size of the wheel depends on the quantity of water to be passed in a given



Radial Section. Günther's Jonval Turbine.

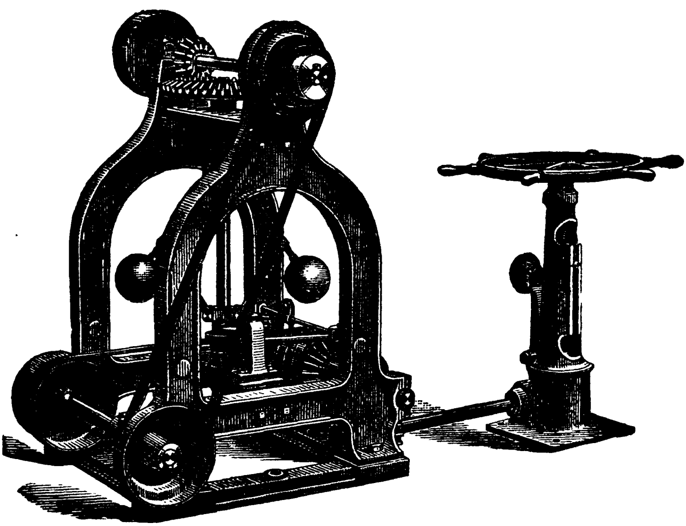


Circumferential Section. Günther's Jonval Turbine.

time. The reason why this class of turbine is not used for small power with high falls is, that very little water would be required, and, therefore, the dimensions of the wheel would be very small — perhaps impracticably small — whilst the speed of rotation would be very great. With the Girard type on the other hand, the efficiency is not reduced by having only one or two of the

buckets in use at one time. We can, therefore, employ as large a wheel as we like, and only use the requisite number of buckets for the required power. This enables us to get a slower speed of rotation which is usually desirable. The Girard type, however, cannot work with a suction tube, and only works well when clear of the tail water. An inch or two of fall, must, therefore, be sacrificed, and this reduces the efficiency with low falls.

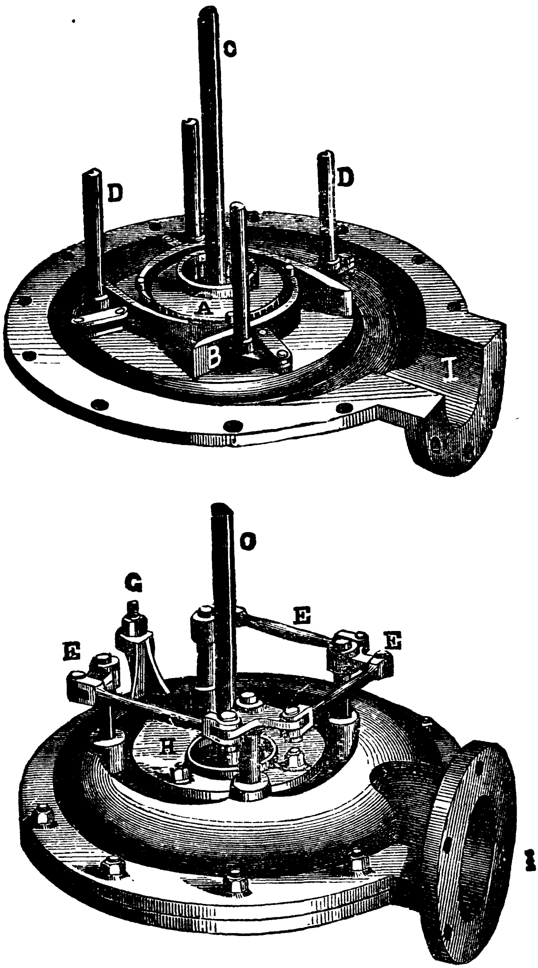
Günther's Turbine Governor.—For many purposes the motion of a turbine is so regular that no automatic control is required, but for some classes of machinery a self-acting governor is desirable. The accompanying figure shows the one made by Messrs. Günther & Sons, of Oldham, for this purpose. It consists of an ordinary



GÜNTHER'S TURBINE GOVERNOR.

Watt governor (see Volume V.) which can shift a belt from a loose pulley to either of two fast pulleys connected by bevel gearing to the pillar for adjusting the turbine. At the normal speed the belt is on the loose pulley, but any change of speed causes the belt to be shifted and the vertical shaft is turned more or less in one way or the other according to the required quantity of water. The governor is driven from a belt on the turbine, or by some shaft from it, and can be disconnected from the hand gear by merely freeing a clutch.

Thomson's Vortex Turbine.—Imagine a wheel placed with its axis coinciding with that of the free vortex already considered in the

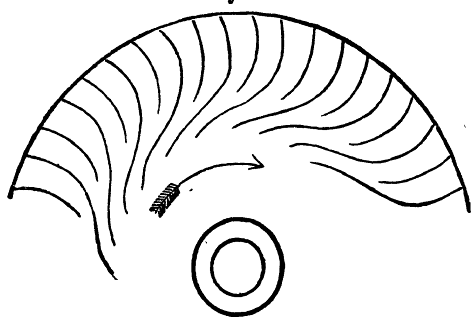


VORTEX TURBINE BY MESSRS. GILBERT GILKES & CO., LTD., KENDAL.

preceding lecture. This wheel will be carried round by the vortex, and if we arrange matters so that the water passes through the

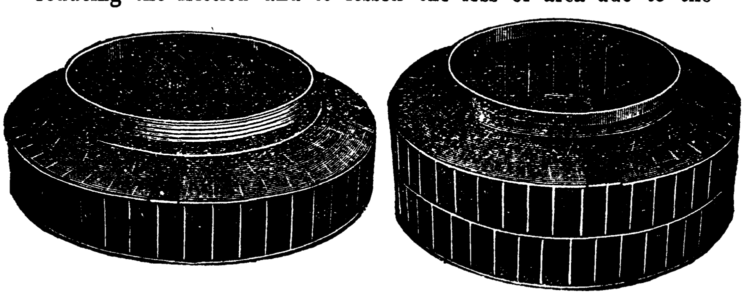
wheel giving up its energy to it while fresh water takes its place, we have the essentials of Professor James Thomson's vortex turbine.

As will be seen from the foregoing illustrations, there are only four guides in the vortex, and by altering the inclination of these, we can adjust the radial component of the motion of the water and the amount of water flowing through the turbine. In the first figure the turbine is represented with its cover removed, and in the second with its case complete. A is the revolving wheel keyed to the shaft C. B is one of the guide blades connected by the bell cranks and shafts D to the outside rods E, which can be adjusted by a screw or by a governor. The shaft runs on a lignum vitæ pivot which is lubricated by the water.



Section of Wheel of Vortex Turbine.

On account of the constrained motion of the water inside the wheel it requires a large number of guides. For the purpose of reducing the friction and to lessen the loss of area due to the

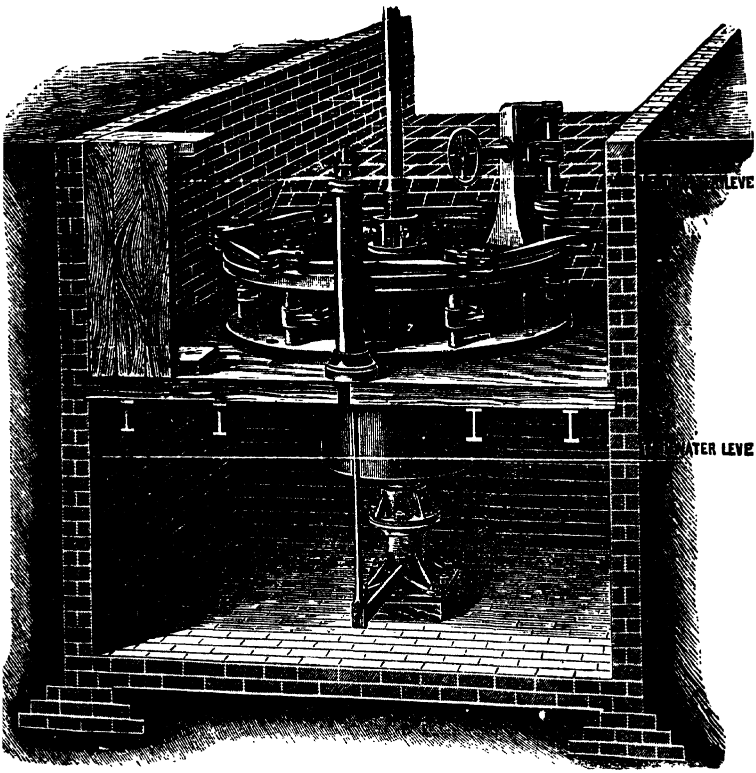


Single and Double Vortex Wheels.

thickness of the guides, it will be seen from the above section that every second guide is only half the length of its neighbouring one. The wheel is made either single or double. In the latter case, it

has the same efficiency at half gate as at full gate, but being of the inward flow "drowned" type it will have a lower efficiency at other loads. Our next figure illustrates one of these turbines as fixed in position.

One great advantage of an inward flow turbine is that, to a certain extent, it is self-governing. When its velocity increases,



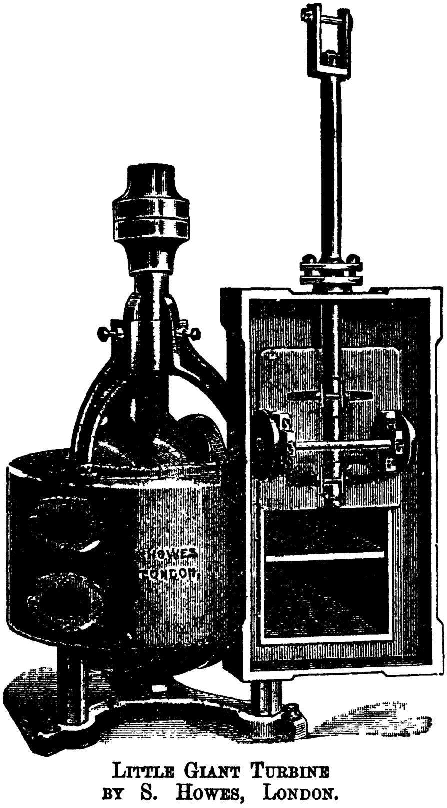
SINGLE VORTEX TURBINE BY GILBERT GILKES & Co., LTD., KENDAL.

so also does the centrifugal force which opposes and consequently reduces the inflow of the water; whilst the opposite action takes place when the turbine is being reduced below its normal speed.

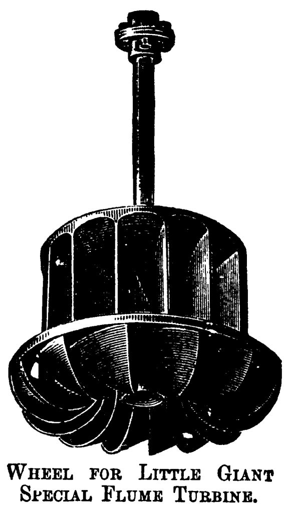
Little Giant Turbine.—We now come to the fourth or mixed flow type of turbine, and as an example we have chosen the "Little Giant" turbine. As may be seen from the illustrations the water

enters at the circumference and passes out at the top and bottom and there is a sluice for regulating the supply. The passage has a division so that water can be entirely shut off from the upper half of the wheel.

The same firm also makes a special "flume" turbine—*i.e.*, one which is placed directly in the water in the same way as the

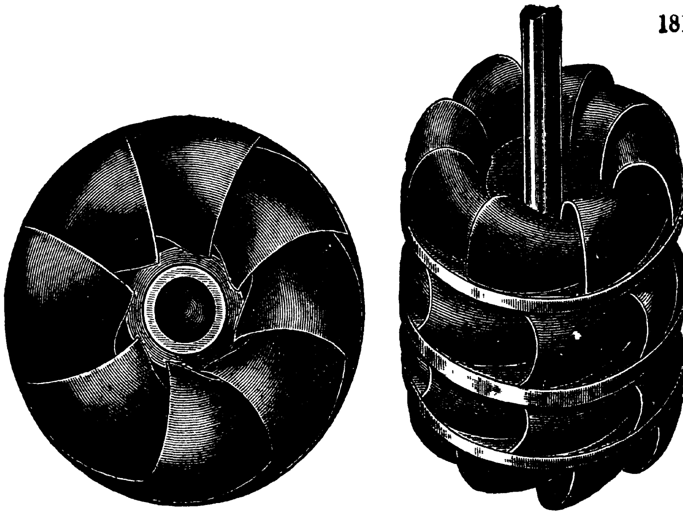


Little Giant Turbine by S. Howes, London.



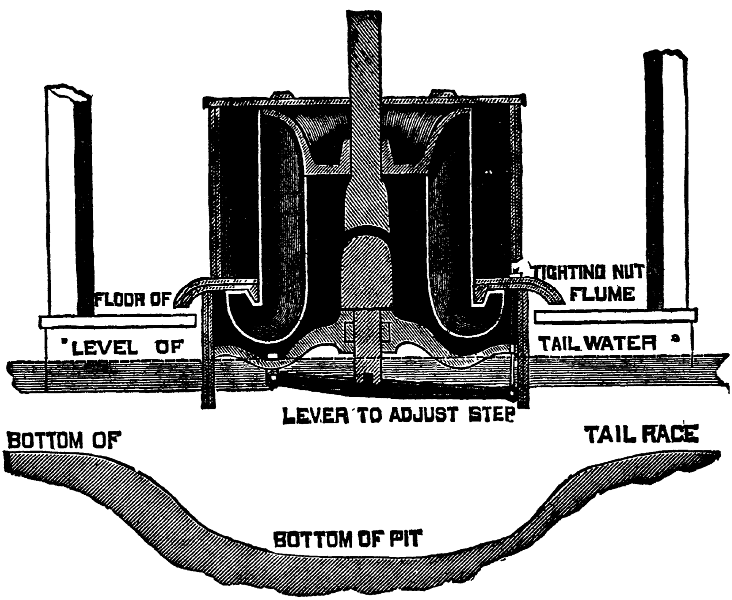
Wheel for Little Giant Special Flume Turbine.

single vortex turbine shown on the previous page. The sectional view shows a turbine formerly made with a wheel similar to that in the Little Giant Flume Turbine, but with a different casing. In this instance, the pivot for the wheel can be raised or lowered by a lever which is clamped in position by a nut. The wheel is held down by a fixed ring fitting in a groove on the wheel, just above the lower splayed-out part.



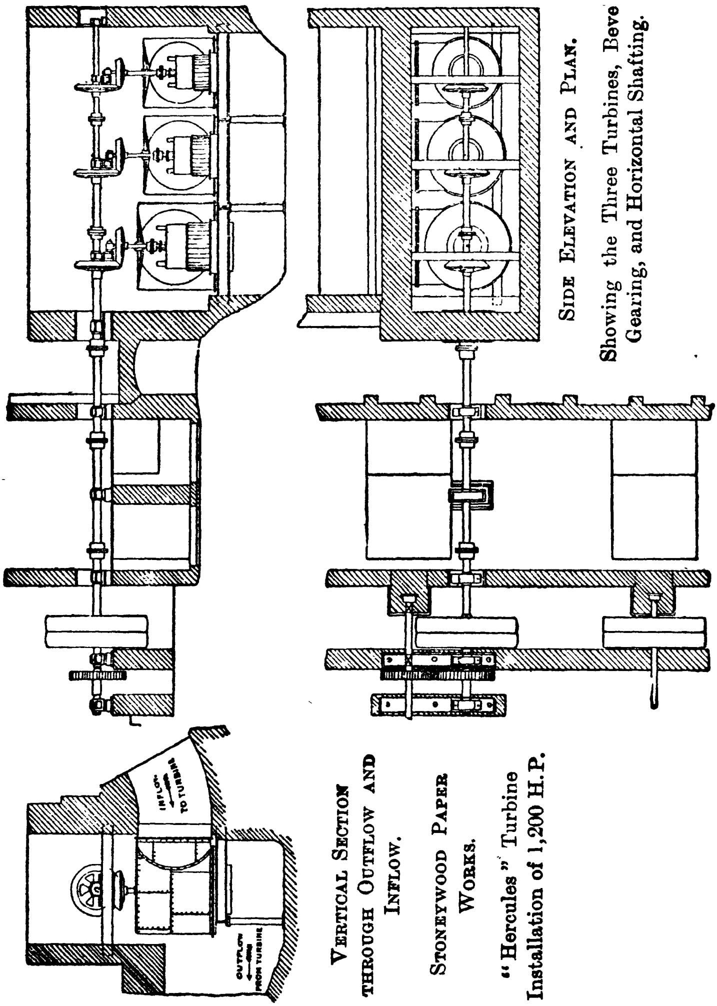
End View. Side View.

Wheel for Little Giant Turbine.



Vertical Section of the Little Giant Turbine.

Stoneywood Paper Works, Turbine Installation.*—This installation consists of three "Hercules" turbines, 54 inches, 48 inches,



SIDE ELEVATION AND PLAN.
Showing the Three Turbines, Bevel Gearing, and Horizontal Shafting.

VERTICAL SECTION THROUGH OUTFLOW AND INFLOW.

STONEYWOOD PAPER WORKS.

"Hercules" Turbine Installation of 1,200 H.P.

* I am indebted to John Turnbull, Jun. & Sons, Glasgow, who put down and set agoing this plant, for the drawings from which the accompanying figures were made.—A.J.

and 42 inches diameter respectively. They are all worked by a fall of 23 feet, and the combined 1,200 H.P. of the three turbines is transmitted by bevel gearing to a horizontal shaft, which revolves at 83 revolutions per minute, as shown by the accompanying views. The power is then conveyed to the various departments of the paper works from this horizontal shaft, principally by belts, but in some cases by gear wheels.

The water is supplied to the turbines through a head race, from the Aberdeenshire river, Don. The sectional area of this race is 22 feet wide by 5 feet 6 inches deep. After passing through the turbines, the water is discharged into a tail race which connects with the river at a lower level.

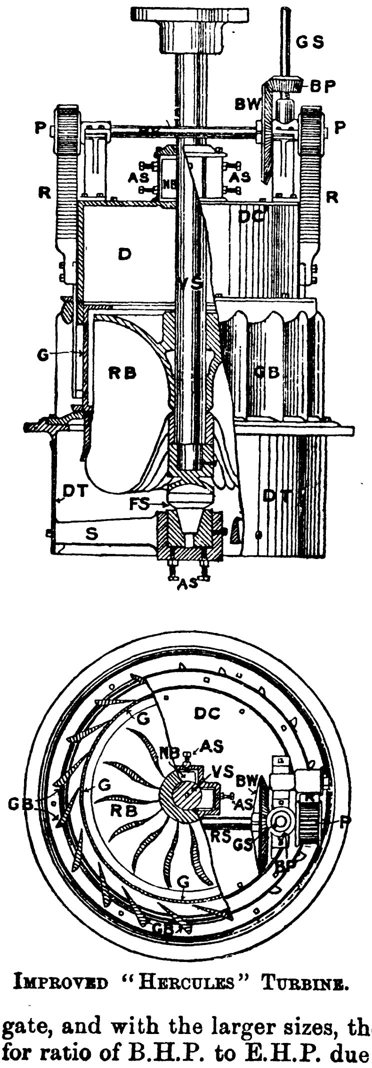
Hercules Turbine.—This low-pressure mixed flow turbine is of American origin and make. Many of these turbines are to be found doing good work in this country under different water heads up to 40 feet. They are made in twenty different graduated sizes, varying from 1 to 2,800 horse, of which the following table shows the capabilities of the former and the latter under the minimum and maximum pressures:—

	Head in Feet.	Horse-Power.	Cubic Feet of Water discharged per Minute.	Revolutions per Minute.
No. 1 Size, . . {	4 40	1·2 35	190 580	266 840
No. 20 Size, . . {	4 40	90 2,830	14,800 46,800	38 117

From the previous and accompanying figures of this turbine, with index to parts, the construction and action will be easily understood. When the gate shaft, G S, is turned in the direction for lifting the gate, it does so through the bevel gear, B P and B W, pinions, P, and racks, R, which are bolted to the upper end of the thin steel cylindrical gate, G. This action permits the water from the source of supply to flow inwards upon the revolving buckets, R B, through the fixed guide blades, G B. Having given up the greater part of its energy to the vertical shaft, V S, the water discharges freely from the draft tube, D T, into and below the surface of the tail race, as shown by the left-hand arrow in the previous figures.

The makers of this turbine and their representatives maintain, that turbines of the "Vortex," and "Little Giant" types, pre-

viously described in this Lecture, cannot possibly give such a high efficiency at part gate as those with fixed guide blades. They also state, that turbines constructed with gates which, when partly closed, alter the angle of the inflowing water as it strikes or presses upon the buckets of the revolver or rotor, are less efficient than the mixed-flow type for the reason, that the angle of direction of the moving water in the latter is not altered at any part of the gate opening.



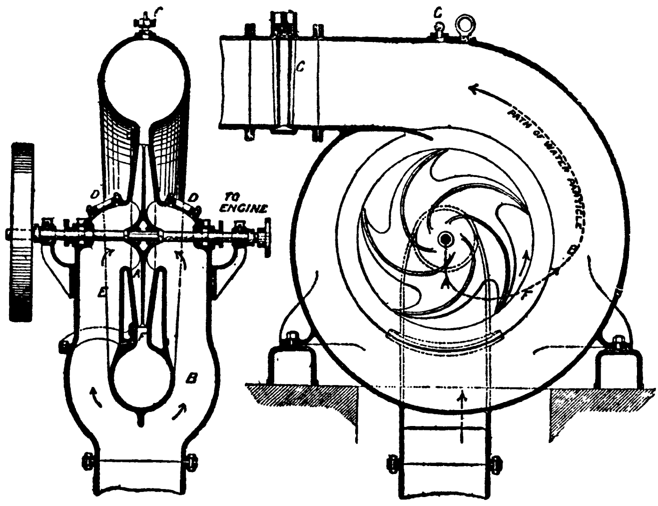
IMPROVED "HERCULES" TURBINE.

INDEX TO PARTS.

G S for Gate Shaft.
B P ,, Bevel Pinion.
B W ,, Bevel Wheel.
R S ,, Rack Shaft.
P ,, Pinions.
R ,, Racks.
G ,, Gate.
G B ,, Guide Blades.
R B ,, Revolving Buckets.
V S ,, Vertical Shaft.
F S ,, Footstep.
S ,, Support for Footstep.
A S ,, Adjusting Screws.
D T ,, Draft Tube.
D ,, Dome.
N B ,, Neck Bush.

Further, that those turbines which are fitted with the cylindrical gate, as herewith illustrated, maintain a higher efficiency at all loads under the maximum power of the turbine than other forms of low-pressure turbines. They guarantee an efficiency of 80 per cent. from half to full gate, and with the larger sizes, they claim 85 per cent. efficiency for ratio of B.H.P. to E.H.P. due to waterhead.

Centrifugal Pumps and Fans.—If certain kinds of turbines be driven by a prime motor the centrifugal force of the fluid carried round with the wheel will cause the fluid to flow outwards in a continuous stream. If the fluid be water the machine is termed a *centrifugal pump*; whereas, if we are dealing with air, it is called a *fan*. Such machines are, therefore, merely reversed turbines; but, in order to get the best results, they must be specially designed, because the best design for a turbine is not by any means the most efficient and suitable for a pump or fan.



CENTRIFUGAL PUMP.

Centrifugal pumps are largely used whenever a quantity of water has to be elevated through a small height, such as in the case of emptying graving docks and sunken vessels, circulating the cooling water through condensers, or dredging sand, gravel, and mud from rivers, whilst centrifrugal fans are employed for ventilating mines, ships, and buildings, as well as for producing an artificial draught to smiths' fires, cupolas, and boiler furnaces, &c.

The illustration* shows a good form of centrifugal pump with curved blades. A is a wheel rotating inside a casing B, thus giving the water which enters at the centre a certain amount of kinetic and pressure energy. F is a small whirlpool chamber which,

* The above figure is reduced from one in Mr. Lineham's *Text-book on Mechanical Engineering* (Chapman Hall & Co.), by kind permission.

as already explained, allows the water to form a free vortex, and converts part of its kinetic energy into energy of pressure. The water moves in a free vortex in the volute-shaped pipe, the path of a partical being shown in the figure by a dotted line and arrows.

The difference of pressure produced by the pump depends upon the density of the fluid in the wheel as well as on the speed of rotation. Consequently, when there is only air in it, the pump is not able to produce a sufficient vacuum to make the water rise into it. In order to get over this difficulty, an ejector G, and a sluice C, are added. When the pump is to be started the sluice is closed and the air exhausted from the pump chamber by a jet of steam being passed through the ejector. The water then rises into the wheel and the sluice is gradually opened as the speed is increased. When the pump is fairly started the steam jet is shut off and the sluice fully opened. Sometimes a non-return valve is placed at the foot of the suction pipe to prevent the pump and pipe emptying when the wheel is stopped. In such a case, the pump is ready to start again at once.

Sometimes centrifugal pumps are made with radial blades. They then require a much larger whirlpool chamber to allow the kinetic energy to change into pressure energy without a serious loss in eddies.

Hydraulic Motors—*Sluice Gates and Stop Valves.*—In most cases it is desirable, if not always actually necessary, to fix a sluice gate or shuttle in the head race as close to the turbine or to the place at which the pipes enter the race as may be convenient, so that in case of need, the water can be entirely shut off from the turbine or pipes. Such shuttles are usually made of timber with iron lifting gear as illustrated by figs. on pp. 162, 163, and 171. For special cases, when timber is unsuitable, it is found advisable to make sluices with the frame, shuttle plate, &c., of iron throughout.

When the water is conducted to a turbine in pipes, it is generally advisable to place an independent stop valve between the turbine and the pipes. This valve may either be of the ordinary full way type or a throttle valve, according to the size of pipes and other considerations.

Supply Pipes.—The necessary diameter of pipe to pass a given quantity of water without excessive loss of head by friction, depends largely upon the total length of the pipe line. It is of the utmost importance that the pipes should not be too small.

The following table has been specially compiled, so that, by reference to it, the necessary size of pipe for any given case can be readily ascertained. This table gives the quantity of

Table giving Quantity of Water passed by Turbine Pipes and Loss of Head by Friction.

Dia. of Pipe in Inches.	N.B.—"Loss of fall" is for every 100 ft. of Piping.	Velocity of Water through Pipes in Feet per Second.								
		2	2·5	3	3·5	4	4·5	5	5·5	6
3	Cub. ft. per min.	5·9	7·3	8·8	10·3	11·8	..	..	..	..
	Loss of fall in ft.	1·1	1·7	2·1	2·9	3·7				
4	Cub. ft. per min.	10·5	13	15·7	18·3	21	..	..	..	..
	Loss of fall in ft.	0·63	1·0	1·4	1·9	2·5				
5	Cub. ft. per min.	16·4	20·4	24·5	28·6	32·7	..	..	..	..
	Loss of fall in ft.	0·47	0·73	1·05	1·4	1·9				
6	Cub. ft. per min.	23·6	29·3	35·2	41·2	47	53	..	..	..
	Loss of fall in ft.	0·37	0·57	0·82	1·1	1·5	1·8			
8	Cub. ft. per min.	42	52	63	73	84	94	..	..	..
	Loss of fall in ft.	0·25	0·39	0·56	0·77	1·0	1·3			
10	Cub. ft. per min.	65	82	98	115	131	147	164	..	..
	Loss of fall in ft.	0·19	0·29	0·42	0·57	0·75	0·95	1·2		
12	Cub. ft. per min.	94	117	141	165	188	212	235	..	..
	Loss of fall in ft.	0·15	0·23	0·33	0·45	0·59	0·75	0·9		
14	Cub. ft. per min.	128	160	192	224	257	288	320	352	..
	Loss of fall in ft.	0·12	0·19	0·27	0·37	0·49	0·61	0·76	0·92	
16	Cub. ft. per min.	168	209	250	293	335	376	418	460	..
	Loss of fall in ft.	0·1	0·16	0·23	0·31	0·41	0·52	0·64	0·77	
18	Cub. ft. per min.	212	264	317	372	425	475	530	580	635
	Loss of fall in ft.	0·09	0·14	0·20	0·27	0·35	0·44	0·55	0·67	0·79
24	Cub. ft. per min.	377	470	565	660	755	845	940	1030	1120
	Loss of fall in ft.	0·06	0·10	0·14	0·19	0·25	0·31	0·38	0·46	0·55
30	Cub. ft. per min.	590	730	880	1030	1180	1320	1470	1610	1760
	Loss of fall in ft.	0·045	0·073	0·10	0·14	0·19	0·24	0·29	0·34	0·42
36	Cub. ft. per min.	850	1060	1270	1480	1700	1900	2120	2330	2540
	Loss of fall in ft.	0·037	0·058	0·084	0·12	0·15	0·19	0·23	0·28	0·34
42	Cub. ft. per min.	1150	1440	1730	2020	2300	2590	2880	3170	3450
	Loss of fall in ft.	0·031	0·048	0·070	0·095	0·12	0·16	0·19	0·23	0·28
48	Cub. ft. per min.	1500	1880	2250	2630	3000	3380	3750	4130	4500
	Loss of fall in ft.	0·026	0·040	0·058	0·078	0·10	0·13	0·16	0·19	0·23
54	Cub. ft. per min.	1906	2382	2859	3335	3811	4288	4765	5241	5718
	Loss of fall in ft.	0·023	0·036	0·052	0·072	0·092	0·117	0·143	0·18	0·21
60	Cub. ft. per min.	2353	2941	3529	4117	4706	5294	5882	6471	7059
	Loss of fall in ft.	0·021	0·032	0·047	0·062	0·081	0·103	0·13	0·16	0·19

water passed by turbine pipes of from 3 to 60 inches in diameter, at velocities varying from 2 to 6 feet per second, together with the frictional loss of fall in feet for every 100 feet of piping. For ordinary cases a velocity of 3 to 4 feet per second may be taken. But, in exceptional instances, such as very high heads, where the loss of a few feet of fall is unimportant compared with the saving in the cost of pipes, a velocity of 5 to 6 feet is often adopted.

If the piping is of considerable length, riveted steel pipes are usually adopted for sizes above 8 inches diameter or thereabouts.

Great care should be taken that the pipes are perfectly clear of rubbish, and, if possible, they should be washed through before making the final connections with the turbine. Otherwise, any debris left in the pipes will be eventually carried into the turbine and cause trouble.

Hydraulic Efficiency of Centrifugal Pumps and Turbines.

To find the "hydraulic efficiency" of the centrifugal pump—*i.e.*, the efficiency if all losses arising from friction and leakage are neglected.

Let T be the torque acting on the pump.

„ ω „ angular velocity of the pump vanes.

„ v_0 „ velocity of whirl perpendicular to the radius at the outlet surface from the vanes (the direction of the flow at the inlet-circle being normal to the latter, there is no velocity of whirl there).

„ V_0, V_i „ linear velocities of the pump vanes at the outlet- and inlet-circles respectively.

„ r_0, r_i „ radii of the opening, in feet, at the outlet- and inlet-circles respectively.

„ Q „ quantity of water in cubic feet per second that the pump is designed to lift.

„ H „ height of the lift in feet.

„ w „ weight in lbs. of a cubic foot of water.

Then, the work done on the pump is $= T\omega$.

Since the velocity of whirl at the inlet from the centre is nil,

$$T = \frac{Qw}{g} v_0 r_0,$$

i.e., the torque equals the moment of momentum communicated to the water passing through the pump per second.

The work done by the pump is $= Q w H$,

therefore hydraulic efficiency of centrifugal pump,

$$\eta = \frac{Q w H}{\frac{Qw}{g} v_0 r_0 \omega}$$

$$= \frac{g H}{v_0 r_0 \omega};$$

or, since $$r_0 \omega = - V_0,$$

hydraulic efficiency of centrifugal pump,

$$\eta = \frac{g H}{v_0 V_0}.$$

The "*hydraulic efficiency*" *of the turbine* can be found in the same way as that of the centrifugal pump.

The work done per second by the water on the wheel is,

$$T\omega = \frac{Qw}{g}(v_i r_i - v_0 r_0)\,\omega \text{ ft.-lbs.}$$

But the effective work done by the water in the turbine is,

$$= \eta w Q H,$$

$$\therefore \qquad \eta w Q H = \frac{Qw}{g}(v_i r_i - v_0 r_0)\,\omega;$$

$$\therefore \qquad \eta = \frac{(v_i r_i - v_0 r_0)\,\omega}{g H}.$$

In order that the efficiency may be as great as possible, v_0 must be 0,

$$\therefore \qquad \eta = \frac{v_i\, r_i\, \omega}{g\,\mathrm{H}}$$

$$= \frac{v_i\, \mathrm{V}_i}{g\,\mathrm{H}}$$

LECTURE V.—QUESTIONS.

1. Sketch an undershot wheel. Explain why its efficiency is so low when it has radial blades, and show how the blades should be made to avoid this loss.

2. Water flows radially at 4 feet per second towards a part of a wheel of a centrifugal pump or turbine which is moving at 12 feet per second, find the angle of the vane that the water may enter without shock. If the vane were radial, at what angle ought the water to be guided so that it might enter without shock; its radial component of velocity being the same as before?

3. Give outline sketches of the common types of water-wheel, and compare their relative advantage.

4. Distinguish between water-wheels and turbines, and explain the advantages of the latter.

5. Sketch the Pelton wheel and describe its action.

6. Sketch and describe a turbine of the Girard type, and mention its advantages and disadvantages.

7. Describe, with sketches, a Jonval turbine, and explain its relative advantages.

8. Sketch the wheel and case of an inward flow turbine for a fall of 50 feet; 8 cubic feet of water per second. Calculate the diameters and breadths of the wheel, the number of revolutions per minute, and the size of the shaft.

9. The vane of a wheel of an internal flow turbine is normal to the rim which moves at 40 feet per second. Water is guided so as to enter the rim with a radial velocity of 4 feet per second; what must be the angle made by the guide blade with the rim, if the water is to enter without shock? If the circumferential area of all the openings of the rim is 1·5 square feet, what volume of water flows per second? We use convenient but not very exact language when we ask—what is the total tangential momentum per second entering the rim?

10. Sketch an inward flow turbine for a fall of 30 feet and 20 cubic feet of water per second. Give its principal dimensions and speed and the angles of guide blades and vanes What is the kinetic energy of a pound of water just entering the wheel? Neglect losses by friction.

11. The radial velocity of water in a centrifugal pump wheel is 2 feet per second. The vane makes an angle of 35° with the circumference; what is the velocity of the water *relatively to the wheel?* The wheel is 2 feet external diameter, and makes 300 revolutions per minute; what is the actual velocity of the water as it leaves the wheel? What is the circumferential momentum of each pound of water? Neglecting radial velocity of water, what is the work done to every pound of water if it enters the wheel with only a radial velocity. and what is the kinetic energy of every pound of water? *Ans.* 3·48 feet per second; 28·63 feet per second; 0·89 lb.-ft.; 27·9 ft.-lbs.

12. Ten cubic feet of water per second enters a turbine wheel with a tangential velocity of 50 feet per second; it enters without shock, the velocity of the rim of the wheel being 50 feet per second; the water leaves the wheel with only a radial velocity. What energy does the water give to the wheel per second?

13. Water, with a radial velocity of 6 feet per second, is really moving along a vane which makes 30° with the circumference of a centrifugal pump wheel, which lets the water leave the wheel with less tangential velocity than that of the wheel itself. Show on a diagram and state the tangential velocity of the water if that of the wheel is 40 feet per second. If the water before it enters the wheel has a radial velocity of 6 feet per second, and enters without shock, what energy is gained per pound of water? What is the gain of kinetic energy? What difference will friction of the passages make in your result?

14. Deduce an expression for the efficiency of a simple hydraulic reaction wheel when you are given the available head and the velocity of the water at the discharging orifice. What will the efficiency be if the head equals 112 feet and the velocity of discharge is 90 feet a second, neglecting friction?

15. Describe, with sketches, a Pelton waterwheel. In what circumstances is such a waterwheel used? What is likely to be the efficiency of the wheel in ordinary working?

16. A Pelton wheel, 2 feet in diameter, runs at 600 revolutions a minute, the available pressure at the nozzle being 200 lbs. per square inch, and the available supply 100 cubic feet per minute. Estimate the available horse-power and the greatest possible hydraulic efficiency of the wheel.

17. Discuss, in general terms, the various losses in a centrifugal pump, pointing out in particular the effect of the whirlpool chamber and of the angle of discharge of the vanes on the efficiency. In a test of a compound centrifugal pump the pressure in the rising main, measured at the pump, was found to be 55·8 lbs. per square inch above atmosphere, and on the suction pipe, measured at the pump, 6·7 lbs. per square inch below atmosphere. The water raised per minute was 119 gallons, whilst the turning moment transmitted to the pump spindle, as measured by a dynamometer, was found to be 27·3 foot-lbs., the revolutions being 1460 per minute. Find the lift, the horse-power of the pump, and the efficiency.

18. Explain what is meant, in referring to hydraulic pressure engines, by the statement that "they contain within themselves their own brakes." It is found that a ram of 10 inches in diameter can lift a load of 20 tons with an uniform velocity of 6 inches per second when supplied with water from an accumulator working at a pressure of 750 lbs. per square inch. Find the coefficient of hydraulic resistance referred to the velocity of the ram, and, assuming this to remain constant at all speeds, find the velocity of the ram when under no load. (*For answer to first part refer to the Froude Water Dynamometer or Brake described in Vol. I.*)

19. Explain, with the aid of sketches, the various devices employed to make water-pressure engines economical when working under loads less than their maximum.

20. The following results were obtained from an undershot water-wheel of 15 feet diameter, 4 feet 6 inches wide, and having only a head of water of 3 feet 6 inches:—

Revs. per Minute.	B.H.P.	Revs. per Minute.	B.H.P.
7·3	6·68	8·9	6·68
8·3	6·7	9·2	6·48
8·4	6·7	10·0	6·4
8·6	6·73	...	...

Plot the circumferential speeds, and the corresponding B.H.P. as co-ordinates. Then, find a constant for the ratio of the velocity of rim of wheel to its diameter at the most effective speed in this case. Also, find the ratio of the speed of rim of wheel to the velocity of the water due to the above head. Can you refer to any reliable data regarding these two ratios or other applicable ratios for water-wheels?

21. Given two turbines with their feed-water pipes identically the same in every respect. Let each turbine be supplied with water from, say, a height of 20 feet above their respective centres. In case (1), the turbine discharges directly into the air; whereas, in case (2) the bottom end of the discharge pipe is led into the tail race below its surface, and the vertical distance from this surface to the centre of the turbine is 20 feet. Which will yield most power, why, and by how much? Sketch and explain your answer. If the centre of the turbine in case (2) were 30 or more feet above the surface of its tail, what would happen, and why?

Lecture V.—I.C.E. Questions.

1. The casing of a centrifugal pump is full of water, but the valve in the delivery pipe is closed; if the diameter of the pump-disc is 12 inches, and it is rotated at 1,200 revolutions per minute, find the pressure at the delivery valve, the length of the suction-pipe being negligible. (I.C.E., *Oct.*, 1903.)

2. Explain the essential differences in working and design of impulse and pressure or reaction turbines. (I.C.E., *Feb.*, 1904.)

3. In an inward-flow turbine, subject to a constant head of water, why is the distribution of pressure in the wheel conducive to steady running when the power required varies, and why does the opposite effect obtain in the case of an outward-flow turbine? (I.C.E., *Feb.*, 1904.)

4. What is meant by the efficiency of a turbine? Explain fully how you would proceed experimentally to find the rate of running that would give the best efficiency for a particular turbine. (I.C.E., *Feb.*, 1904.)

5. Sketch and explain the construction of a centrifugal pump, and point out the means taken to diminish the losses of energy. (I.C.E., *Oct.*, 1904.)

6. An inward-flow turbine wheel works under a head of 60 feet, and makes 380 revolutions per minute. The diameter of the outer circumference of the wheel is 24 inches and of the inner circumference 12 inches. The velocity of the water entering the wheel is 44 feet per second, and the angle it makes with tangent to the wheel is 10 degrees. Assuming the velocity of flow through the wheel to be constant, determine graphically the direction of the tangent to the vane of the wheel at the inlet and outlet. Sketch a suitable form for the vane. Determine the hydraulic efficiency of the turbine. (I.C.E., *Oct.*, 1904.)

7. Show that the efficiency of a Pelton wheel is a maximum—neglecting frictional and other losses—when the velocity of the cups equals half the velocity of the jet. 25 cubic feet of water are supplied per second to a Pelton wheel through a nozzle, the area of which is 44 square inches. The velocity of the cups is 41 feet per second. Determine the H.P. of the wheel, taking a reasonable value for the efficiency. (I.C.E., *Feb.*, 1905.)

8. Prove, that if a mass of water enter the wheel of a centrifugal pump with no component tangential to the wheel, the theoretical height to which the water will be lifted is $h = \frac{V v}{g}$, where V equals the velocity of the outer periphery, and v the component of the velocity of the water when it leaves the wheel along the tangent to the wheel. If V is 60 feet per second, and the radial velocity of flow 5 feet per second, and the angle the tangent to the vane makes with the direction of motion is 120 degrees, determine h, and also the pressure-head at the point where the water leaves the wheel. (I.C.E., *Feb.*, 1905.)

9. A B and A C are two lines inclined at 30 degrees. A jet of water moves in the direction A C with a velocity of 24 feet per second, and a vane in the direction A B with a velocity of 12 feet per second. Show how to find the form of a vane so that the water may come on it tangentially, and leave it in a direction perpendicular to the direction of motion of the vane. Determine the pressure on the vane in the direction of motion due to each pound of water striking the vane. (I.C.E., *Feb.*, 1905.)

10. A reaction wheel is revolving with a given velocity V, and from its orifices water is issuing with a given velocity v. Show that for each lb. of water the useful work $= \frac{V(v - V)}{g}$, and the efficiency $= \frac{2V}{v + V}$. (I.C.E., *Oct.*, 1905.)

11. A Pelton wheel 3 feet diameter revolves at the rate of 200 revolutions per minute, the velocity of the jet of water impinging on the buckets being 80 feet per second. Find the hydraulic efficiency of the wheel. What should its velocity be for the efficiency to be a maximum? (I.C.E., *Feb.*, 1906.)

12. Define impulse and reaction turbines. What is the efficiency of a turbine delivering 5,000 horse-power when 193,620 gallons of water per minute pass through it under a head of 136 feet? (I.C.E., *Oct.*, 1906.)

13. Water flows through a circular penstock, 30 inches in diameter, at a mean velocity of 50 feet per minute. Find the head of water at the centre of the opening when the discharge is 450 gallons per minute. (I.C.E., *Feb.*, 1907.)

14. (*a*) Sketch an end view or vertical section of an overshot water-wheel, indicating the head and tail races, total head, entrance fall, exit fall, and the angles of entry and exit. (*b*) State the two fundamental requirements necessary to secure high efficiency. (I.C.E., *Feb.*, 1907.)

15. (*a*) Find the theoretical H.P. of an overshot water-wheel, 22 feet diameter, using 2,000,000 gallons of water per 24 hours under a total head of 25 feet. (*b*) Express the hydraulic efficiency of an overshot wheel in terms of head, entrance fall, and exit fall. (I.C.E., *Feb.*, 1907.)

16. Make sketches showing the construction of a centrifugal pump. Explain the principal causes of loss of energy in the centrifugal pump, and show how in practice attempts are made to diminish these losses. (I.C.E., *Oct.*, 1907.)

17. A quantity of water, Q cubic feet per second, flows through a turbine, the initial and final directions of flow being known. Find the couple exerted on the turbine, and hence, when the angular velocity of the turbine is known, the work done on the turbine. If h is the total difference of level between the head and tail water of the turbine, u the velocity with which the water leaves the turbine, and the hydraulic efficiency of the turbine is η, show that the total fraction of the head lost in the supply pipe, turbine wheel and casing, and delivery pipe, is

$$k = 1 - \eta - \frac{u^2}{2gh}.$$ (I.C.E., *Oct.*, 1907.)

18. The water approaching a water-fall having a drop of 100 feet is measured by a weir 20 feet long; the head over the weir is observed to be 9 inches. If the water, instead of being allowed to pass over the fall, is led into a penstock and used to drive turbines, and 65 per cent. of the available energy is converted into useful work, find the horse-power of the turbines. The coefficient for the weir is to be taken as 3·33. (I.C.E., *Feb.*, 1908.)

19. It is desirable that the turbines of Question 6 should be placed at least 15 feet above the bottom of the fall. Show, by sketches, the kind of turbine you would use, explaining carefully how full advantage is taken of the available head. If the velocity with which the water from the turbine enters the tail race is 10 feet per second and other losses are neglected, find the efficiency of the turbine. (I.C.E., *Feb.*, 1908.)

20. Describe, by aid of sketches, either (i.) the principal features of a hydraulic power-station for supplying water at 700 lbs. pressure per square inch to a large city, or (ii.) a hydraulic installation for utilising by means of turbines the natural fall of a river of about 40 feet. (I.C.E., *Oct.*, 1908.)

21. A jet of water 2 square inches in area and at a velocity of V feet per second impinges upon a set of hemispherical vanes, and drives them in its

own direction at a velocity of v feet per second. Deduce an expression for the work done, and plot a curve showing its amount for all possible speeds of the vanes when V is 400 feet per second. Describe a modification of the hemispherical vane used in a Pelton wheel, and show what are the practical reasons for the departure from the hemispherical form.

(I.C.E., *Oct.*, 1908.)

22. Describe by aid of sketches the construction of one form of rotary hydraulic engine operated by water at a high pressure, and explain what circumstances limit the speed of such an engine, and what causes may lead to a reversal of stress in the engine. (I.C.E., *Oct.*, 1908.)

23. Describe the principal considerations which you would take into account in the design of a turbine suitable for a low fall liable to fluctuations of level and water-supply. Describe the construction and special features of advantage of a form of turbine which you consider particularly well adapted to give a high all-round efficiency with a water-supply which is normally 6,000 cubic feet per minute at 10 feet head rising to 9,000 cubic feet per minute at a head of 6 feet in times of flood. (I.C.E., *Oct.*, 1908.)

24. Describe and illustrate, by sketches in good proportion and detail, the construction and action of a high lift centrifugal pump capable of lifting water against a head of not less than 300 feet. Explain what are the essential features of difference between the form you describe and that of a low lift centrifugal pump designed for a head of 30 feet.

(I.C.E., *Oct.*, 1908.)

ANSWERS TO I.C.E. QUESTIONS.

1. Pressure at delivery valve = 26·5 lbs. per square inch.

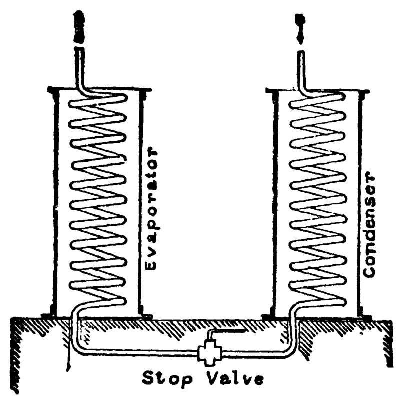
6. Direction of tangent to the vane of wheel at inlet = 113° 6′; direction of tangent to the vane of wheel at outlet = 20° 54′; hydraulic efficiency of turbine = $66\frac{2}{3}$ per cent.

7. H.P. of Pelton wheel = 235·8 H.P., taking an efficiency of 80 per cent.

8. h = 106·4 feet; the pressure head at point where the water leaves the wheel = 55·4 feet.

9. The vane makes at inlet an angle 126° 12′ with A B; the vane makes at outlet an angle 45° with A B; pressure on vane in the direction of motion due to each 1 lb. of water striking it = ·275 lb.

can take place. If we take two strong coils of piping and surround each with a vessel of water and then connect the two by means of a stop valve at their lower ends, as shown by the accompanying figure, we shall have a simple form of refrigerating machine. Suppose, that when the stop valve is closed, we charge the condenser coil with gas under liquefying pressure, by means of a force pump. The water surrounding the coil will absorb the heat which has been imparted to the gas by compression, and the condensed liquid will gradually accumulate at the bottom of the coil. On opening the stop valve, this liquid will run into the second or evaporating coil, and the pressure here being lower than is necessary for maintaining the liquid state of the material, evaporation will at once commence. The heat necessary for evaporating this liquid is

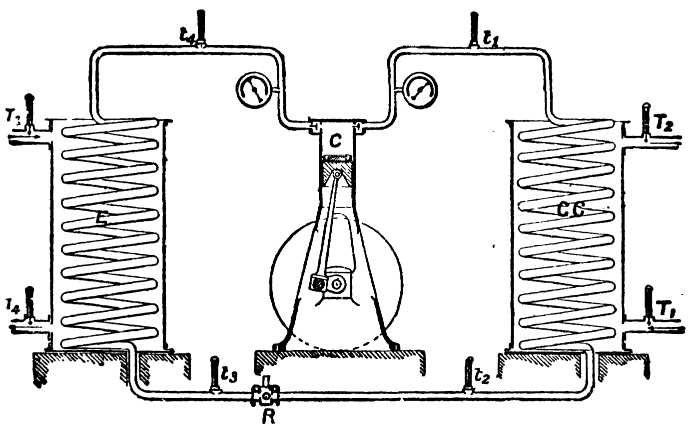


ELEMENTARY REFRIGERATING MACHINE.

absorbed from the water surrounding the evaporating coil, which will thereby become considerably reduced in temperature. To accomplish this result in practice three things are necessary :—(1) A compressor, to raise the pressure of the gas to whatever may be necessary for its liquefaction; (2) a surface condenser, to remove the heat generated by the mechanical work of compression; (3) an evaporating vessel, where the liquid may re-evaporate into a gas, and absorb heat in the operation.

Simple Refrigerating Machine.—The following figure of a simple refrigerating machine will explain the cycle of operations. C is a compressing pump delivering gas under pressure into the condensing coil C C, which consists of a strong worm of iron or copper piping immersed in a tank of water. R is a

regulating-stop valve having a fine adjustment. E is the evaporator which consists of a coil of piping similar to the condensing coils. It is also immersed in a tank containing the water or other fluid to be cooled. The regulator R is closed, as soon as the pressure in the condenser has risen to that at which liquefaction can take place and the gas commences to condense on the inner surface of the coil C C. The drops of liquid descend and accumulate in the lower portion of this coil. The regulator is then opened, with the result, that a small quantity of liquid escapes into the evaporator. Now, since the compressor draws its supply of gas from the evaporator, the pressure in the evaporator must be less than in the condenser. Consequently,



SECTIONAL DIAGRAM OF A SIMPLE REFRIGERATING MACHINE.

INDEX TO PARTS.

C for Compressor.	E for Evaporator.
C C „ Condensing Coils.	t_1 to t_4 „ Thermometers.
R „ Regulating Valve.	T_1 to T_4 „ Thermometers.

the liquid commences to boil, and absorbs heat for its transformation into a gas from the surrounding liquid. The temperature of this liquid is therefore naturally reduced by the operation. The liquid within the coil is entirely re-converted into a gas which ultimately finds its way to the compressor, and thus the cycle of operations is completed.

Suppose four thermometers be inserted into the pipes conveying the gas to and from the condenser and evaporator, as shown at t_1, t_2, t_3, t_4. It will be found that they do not register alike, for t_1 will show the highest temperature, then t_2 and t_3 will be

some degrees lower, and t_4 will be lowest of all. Suppose now, that the liquids in the vessels surrounding C C and E be caused to circulate in the direction of the arrows, and that thermometers T_1, T_2, T_3, T_4, be placed on each of the inlet and outlet pipes. It will be found, that the temperature of the incoming water T_1 is lower than T_2 the temperature of the water going out of the condenser; also, that T_3 the temperature of the liquid entering the evaporator is *higher* than T_4 its temperature as it leaves this vessel. This shows, that with respect to the gas or liquid within the coils of the condenser or evaporator, heat is lost in the condenser and gained in the evaporator. The amount of the former is represented by the difference between T_1 and T_2 multiplied by the weight of water passed through the condenser in pounds, and the latter may be expressed in terms of the difference between T_3 and T_4 multiplied by the weight of the fluid passing through the evaporator, and by the specific heat of this fluid.

If we could construct an ideal machine, in which the liquefaction of the gas was automatic, it would be found that the loss of heat in the condenser, measured in thermal units, was exactly equal to the gain of heat in the evaporator. The sensible heat gained and lost by the fluids surrounding the coils in the condenser and evaporator respectively, would be the exact measure of the latent heat of the refrigerating medium, as abstracted in the condenser and returned in the evaporator. It is, however, necessary to change the physical condition of the gas between the evaporator and condenser, so that it can be liquefied in its passage through the latter vessel. Suppose that the pressures in both evaporator and condenser are the same and constant. In order to ensure condensation and liquefaction in the condenser, its temperature would have to be constantly maintained *below* that of the evaporator, a condition of things which is manifestly impracticable since the evaporator is becoming colder with every repetition of the cycle of operations. This difficulty must therefore be met in another way. If we wish to liquefy any gas, it is necessary to bring its molecules closer together, and this can be accomplished either by *increasing* the pressure or by *decreasing* the temperature of the gas, or both. Now, since it is not in this case practicable to reduce the temperature, the only alternative is to raise the pressure by means of the pump already referred to, which draws the gas from the evaporator and delivers it at an increased pressure into the coils of the condenser. But in order to compress a gas, mechanical work must be performed upon it, and this work re-appears in the form of heat. The temperature of the gas after compression is

therefore considerably higher than it was at the lower pressure on leaving the evaporator. This heat, in addition to the heat imparted in the evaporator, has to be abstracted and carried away by the cooling action of the water of the condenser.

As stated at the beginning of this Lecture, if we convert a unit weight of any liquid into a gas, we require the addition of a definite amount of heat, and to reconvert this gas into a liquid we require the abstraction of the same amount of heat, the amount being constant for any one liquid at a constant pressure and temperature. All gases do not require the same expenditure of energy to raise them to the same pressure, because they vary in what may be called their compressibility, and some gases occupy a smaller volume than others after an equal amount of compression. Carbon dioxide, for example, according to Regnault, only requires about 75 per cent. of the work necessary to produce the same amount of compression, as air or hydrogen. We can, by experiment, readily determine the exact pressure at which liquefaction will take place at any temperature; and knowing this, the machine can be designed of suitable strength to withstand the necessary pressure.

Owing to the difference in the power required to increase the pressure of different gases, it follows that the amount of heat imparted during compression must with some gases be greater than with others. This fact is of great importance in the selection of a suitable gas, and particularly so if cooling water be scarce. But whatever gas be employed, the pressure necessary to liquefy it must always be increased to a greater or less extent as the temperature of the cooling water rises.

Having considered the principles upon which an evaporative refrigerating machine depends for its action, we are now in a position to examine into the actual question of the interchangeability of heat and work. We can moreover at once establish a coefficient of efficiency for any refrigerating material.

Let, L = Latent heat of evaporation of the refrigerating medium in B.Th.U.

And, H = Heat imparted during compression in B.Th.U.

It follows, from what has been said, that the coefficient of efficiency will be $L \div H$, and, neglecting external losses, wL will represent the heat abstracted in the evaporator, whilst $H + wL$ will equal the heat added to the cooling water of the condenser, where w is the weight in lbs. of the gas entering the condenser in a given time. It is, of course, impossible that a machine could work under such ideal conditions as we have assumed, since there must always be the effect of the high or

low temperature of surrounding bodies to determine whether there will be a loss or gain of heat in one or other of the parts of the machine. For instance, it is almost certain that the evaporator will be colder than the atmosphere; in that case, no matter how carefully it may be insulated, there will be some conduction of heat and the net quantity of the heat abstracted from the liquid to be cooled can only be $(w\,L - x)$, where x, is the amount of heat derived from outside sources. The amount of heat imparted to the cooling water in the condenser will therefore be $H + w\,L \pm y$; where y, is the heat lost or gained in the condenser due to the difference in temperature between it and its surroundings.

There is still another correction to be made to the above formula. When, as is usual, the evaporator is maintained at a very low temperature, a certain amount of heat must be imparted to it by the refrigerating liquid itself, as it is entering the evaporator in a comparatively warm condition. Thus, supposing there be t degrees difference in temperature between the condenser and evaporator, a unit weight of the refrigerating liquid will as it were import into the evaporator $t\,s$ thermal units; where s is the specific heat of the liquid in question. Therefore, if W be the weight of refrigerating liquid passing into the evaporator in a given time, the heat abstracted in the evaporator will be represented by the expression:—

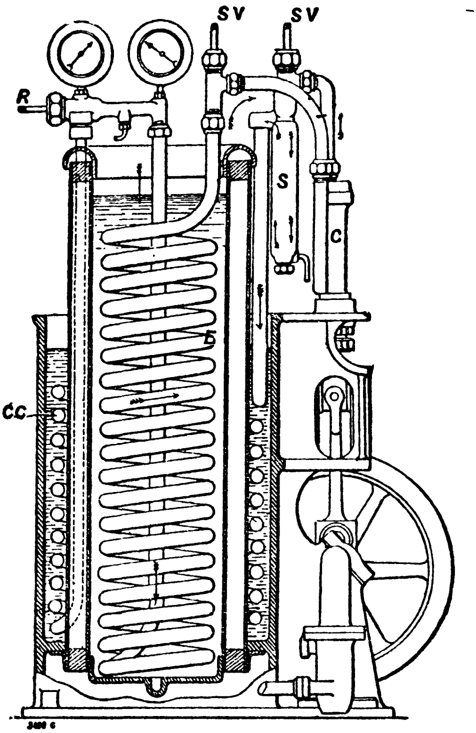
$$w\,L - x - W\,t\,s.$$

Of course, $W\,t\,s$ will not in practice amount to a great deal; but, as Professor Linde has pointed out, it must not be neglected in an exact calculation of the work performed by any refrigerating machine. If there be no leakage, then on the average W will be the same as w.

These formulæ cannot be applied with absolute certainty in practice, owing to the impossibility of making all the necessary corrections due to the gain or loss of heat in the various parts of the machine, and owing to the friction of the gas in constricted passages. But, with care, this gain and loss of heat can be very nearly accounted for in an ordinary machine, as manufactured for commercial purposes and working under the conditions of everyday practice.

Carbon Dioxide Refrigerating Plant.—One method of cooling buildings, &c., on a large scale, is to employ a strong brine obtained by dissolving sodium chloride or common salt in water. This brine is first cooled by passing it around the evaporator of a refrigerating machine, and then circulating it in pipes placed within the chambers which it is desired to cool.

In the accompanying figure all the essential parts are shown of a small refrigerating machine as manufactured by Messrs. J. & E. Hall, of Dartford, for cooling small provision stores, dairies, &c., where the pump may be conveniently driven by a belt. In larger machines, the evaporator is contained in a separate



HALL'S CARBON DIOXIDE REFRIGERATING PLANT.

INDEX TO PARTS.

C for	Compressor.	G for	Gauges.
S ,,	Separator.	E ,,	Evaporator.
CC ,,	Condensing Coils.	SV ,,	Stop Valves.
R ,,	Regulator.		

vessel and the compressor is driven by a compound- or triple-expansion engine; but, for the sake of compactness in this case, the evaporator is placed within its condenser, and the intervening space between them is carefully insulated by means of some non-conductor, such as hair felt or slagwool. The coils of

piping which form the condenser and evaporator are welded into continuous lengths and so connected that all joints shall occur in accessible positions. These and all the other gas joints are made by inserting copper rings turned from the solid metal between a pair of flanges or union coupling. This form of joint has been found very satisfactory. The condenser casing, Corliss frame and bearings for the compression pump, &c., are all made of cast iron in one casting. The compressor C is made of special hard bronze in order to ensure freedom from spongy places, while the suction and delivery valves are identical in shape and size so that they may be interchangeable. The compressors for larger machines are bored out of solid steel forgings. This ensures strength together with sound material. A true bore is also provided for the smooth working of the cupleathers with which the pistons are packed. The gland is made gas-tight by means of two U-leathers fitted over the compressor-rod and glycerine is forced between them under a somewhat greater pressure than that in the compressor. Any leakage which takes place is therefore of glycerine—outwards (which can be collected and used over again) and inwards—which both lubricates the interior of the compressor and fills up the clearance spaces, thereby increasing the efficiency of the machine. The superior pressure of glycerine in the gland is obtained by utilising the pressure in the condenser acting through a small intensifier, similar to those in use in hydraulic installations. Any glycerine which passes into the compresser, beyond what is necessary to fill up the clearance spaces is discharged with the gas through the delivery valves. In order to prevent this glycerine passing into the condenser coils, all the gas is delivered into a separator S and caused to impinge against the sides of this vessel. The glycerine adheres to its sides and drains to the bottom from which it may be drawn off from time to time, thus permitting the dry compressed gas to pass away by an opening at the top of the separator to the condensing coils.

One feature of these machines is the safety valve, which is fitted to the gas circuit immediately above the compressor, so that no harm can be done to the machine even if carelessly started with the stop valves closed. It consists of an ordinary spring safety valve, beneath which is a thin copper disc, designed to burst at a certain pressure. This disc can be made perfectly gas-tight, which could not be so easily accomplished by the spring safety valve alone. The latter only comes into play in the event of a rupture of the copper disc.

Anhydrous Ammonia as a Refrigerating Agent.—The most important advantages possessed by anhydrous ammonia as an

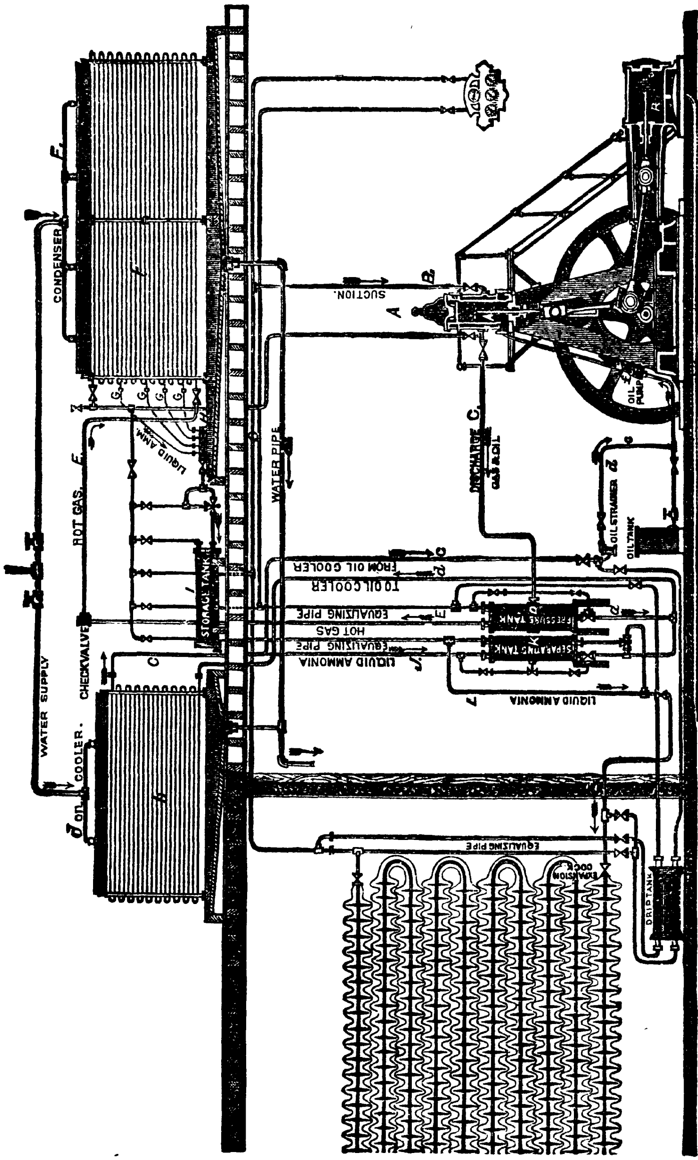
agent for cooling purposes are, its freedom from the danger of explosion, its great latent heat and low pressure of vaporisation.* The latent heat of vaporisation of 1 lb. of carbonic acid at 0° F. is 123 units, and of ammonia 555 units, while the respective pressures in lbs. per square inch, at the same temperature, are 310 lbs. in the case of carbonic acid and only 30 lbs. with ammonia. It follows, therefore, that a carbonic acid plant must be constructed to deal with pressures of about 1,000 lbs. per square inch as against only 150 lbs. or so, in an ammonia machine. The exact pressures in each case are directly proportional to the temperature of the condensing water.

As already stated, this agent is more commonly used than any other, and in the United States of America, where refrigeration is applied to an extent unknown elsewhere, the machine generally employed is on the ammonia compression principle. It is similar to the carbonic acid machine in so far as the complete system consists of (1) a compression, (2) a condensing, and (3) an expansion part; moreover, the cycle of operations is exactly the same.

De La Vergne's Refrigerating Plant.—There are many different kinds of ammonia machines in use, but a general description of one of the best known and most extensively applied—viz., the "De La Vergne" as manufactured by Messrs. L. Sterne & Co., Ltd., of the Crown Iron Works, Glasgow, may be taken as a typical example.

In the following figure, A represents the ammonia compressor driven by a steam engine R. The gas which is returned from the expansion coil N, placed in the cooling chamber, enters the cylinder A by the pipe B, and after being compressed therein it is discharged, through the pipe C into a pressure tank D, together with a certain amount of sealing oil. Here, the oil, being heavier than the ammonia gas, naturally falls to the bottom, and the hot ammonia passes from the top of this tank by a pipe E to the condensers F; where, the cooling action of cold water trickling over the pipes causes the gas to liquefy. It then passes through pipes G, G to a header H, and from thence, to a storage tank I, which is simply a receptacle for holding a reserve supply of liquid ammonia. From this tank

* A liquid with a high latent heat of evaporation need not necessarily be a good refrigerating agent, and *vice-versâ*. What is required is, that its specific heat should be low in proportion to the latent heat of evaporation. Or, we require as great a *difference* as possible between the latent heat of evaporation and the specific heat of the liquid multiplied by the range of temperature in the condenser and refrigerator.

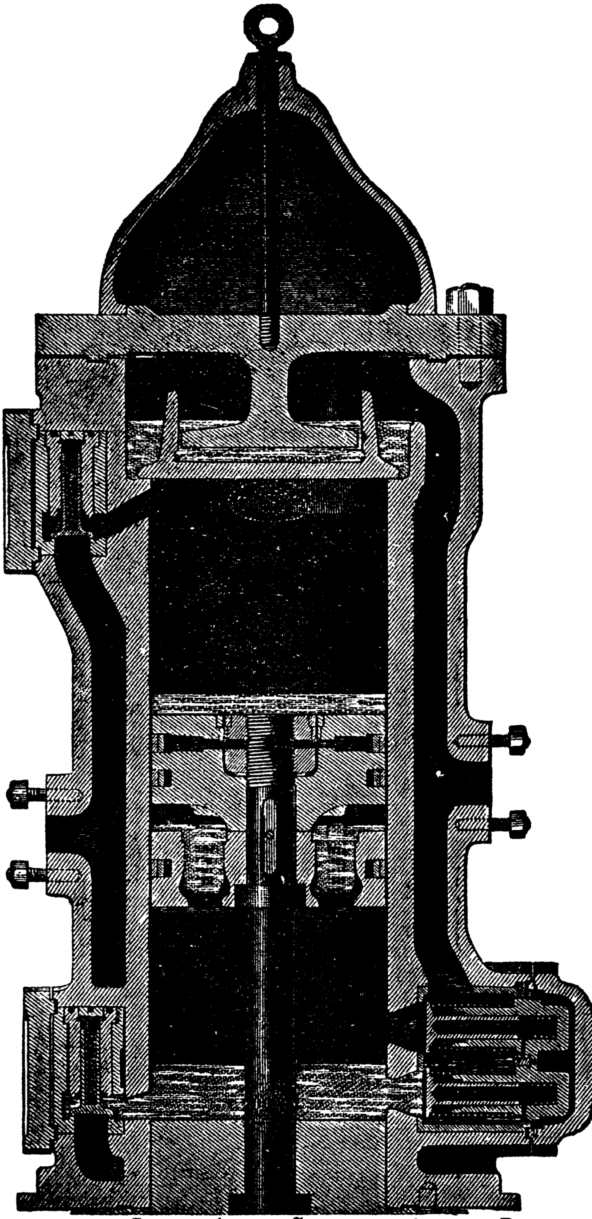


De La Vergne Refrigerating Plant. By L. Sterne & Co., Crown Iron Works, Glasgow.

it is conveyed by a pipe J to a separating vessel K, where any particles of oil that may have been carried over with the liquid are finally separated and the pure liquid ammonia is free to leave it by a pipe L to the expansion coils in the chambers to be cooled. The cock for admitting the ammonia to these coils can be regulated to any degree of minuteness. It thus serves to separate the high pressure from the low pressure part of the apparatus. Hence, the liquid ammonia on passing the expansion cock enters the cooling coils, which are maintained at a low pressure by the pumping action of the compressor. Here, it immediately flashes into gas and by abstracting from its surroundings the heat necessary to cause this change, the temperature of the room is lowered to any desired extent. After having thus done its work in the cooling chamber, the gas is returned to the compressor by a pipe B, to again undergo the same cycle of operations.

The sealing oil passes from the bottoms of the pressure and separating tanks D and K, by the pipes *a* and *d* to the oil cooler *b*; thence, by pipe *c* to the oil strainer *d* and the pipe *e* to the oil pump *f*; by which, it is again circulated through the compressor A.

De La Vergne's Double-Acting Compressor.—The accompanying figure is a section through a "De La Vergne" double-acting compressor, and shows the use of the oil seal. In all ammonia compressors, a certain amount of oil is required for lubricating purposes, and if the compressor be arranged in the ordinary way, the discharge valves at the lower end are placed either on the bottom or at the side, with the result that the oil is discharged *before* the gas. The oil ought, however, to be discharged *after* all the gas is gone; otherwise, re-expansion takes place which would entail a loss of efficiency. In the "De La Vergne" compressor this difficulty has been avoided in the following manner:—At the lower right-hand end of the compressor, two discharge valves are fitted into a side pocket, with the one fair above the other. On the down stroke, either of the valves or both may open until the piston covers the upper one, when only the lower valve is open to the condenser. In the further course of the piston and as soon as the lower valve is also closed, the upper one comes into direct communication with an annular chamber in the piston. This chamber has valves in its bottom side which open into it, as soon as all other inlets on the lower side of the piston are closed. The gas, therefore, first leaves the compressor and then the oil follows, thus permitting no gas to remain in the lower side after the completion of the down stroke. The effect of the oil seal is to make the compressor



DE LA VERGNE'S DOUBLE-ACTING COMPRESSOR FOR HIS REFRIGERATOR.

work with practically no clearance and thus a maximum of efficiency is obtained. The oil also serves to carry away a considerable amount of the heat of compression and to seal all the valves and stuffing-boxes.

Attention may now be drawn to a few of the details of the above plant. In the first place, it will be noticed that the ammonia condenser is not of the ordinary type where the coils containing the gas are usually submerged in a water tank, but they are of the open or atmospheric type. Here, water is kept constantly trickling over the condenser pipes, and the cooling action is therefore considerably assisted by the evaporation thereof from the surface of the pipes, which enables a maximum of condensation to be effected with a minimum of water supply. It also leaves all the pipes of the condenser open for examination and cleansing. This style of condenser is now coming into extensive use for the condensation of steam in large factories. In the second place, it will be seen that the refrigerating or cooling effect is caused by the direct expansion of the ammonia in pipes placed in the chamber to be cooled. This does away with the unavoidable loss of efficiency due to the use of a supplementary medium such as brine. It, however, necessitates very careful coupling up and jointing all the expansion coils, in order to prevent any leakage of the ammonia gas; more especially, in the case of a large plant where there may be as much as ten or more miles of piping in these cooling coils. In practice, however, these details have been so carefully worked out, that many hundreds of miles of such piping are constantly at work without giving the slightest trouble. Consequently, the old-fashioned method of brine circulation is not now so generally employed except on board ship, where there is a possibility of undue rocking or straining of the pipes and where it is considered advisable to use something that would cause no disagreeable odour in case of a broken pipe or joint.

In applying the refrigerating machine to the manufacture of ice, the simplest method is to place the expansion or cooling coils in a tank filled with brine or other non-congealable liquid, while the water to be frozen is placed in moulds of suitable size, which are then inserted into this brine until frozen. The purpose served by the brine in this case is to convey the heat from the water to the cooling pipes. It is therefore generally kept in slow circulation in order to ensure that the temperature shall be as uniform as possible throughout the tank. If ordinary well water be placed in the moulds, the resulting ice will contain so much air that it will be turned out of a milky white and opaque colour; but if the water whilst in the process of freezing be kept

in slow motion by means of agitators, this air escapes, and a clear glassy ice is the result.

Another method of obtaining clear ice is to use distilled water. This has also the advantage of getting rid of any objectionable matter which might be in solution.

In modern refrigerating plants, electric motor drive is commonly employed, the motor being coupled to rotary compressors by means of which the necessary pressure is obtained. This construction is used particularly in the case of small installations.

The Linde System of Refrigeration.—This system was first introduced into Germany in the year 1875 by Professor Linde, who was then one of the teaching staff at Munich University. In this country, however, prior to 1888, the principal cold-producing machinery, as manufactured for both land and marine purposes, was the simple cold air machine, in which refrigeration is produced by the compression, cooling when under compression by means of water and subsequent expansion of ordinary atmospheric air. These machines, although simple in construction and giving very good results, possess the disadvantage of requiring a large amount of power to work them in comparison with those employing more efficient refrigerating agents. Consequently, the former method has now very largely given way to one or other of the latter, of which the Linde system is one of the most successful.

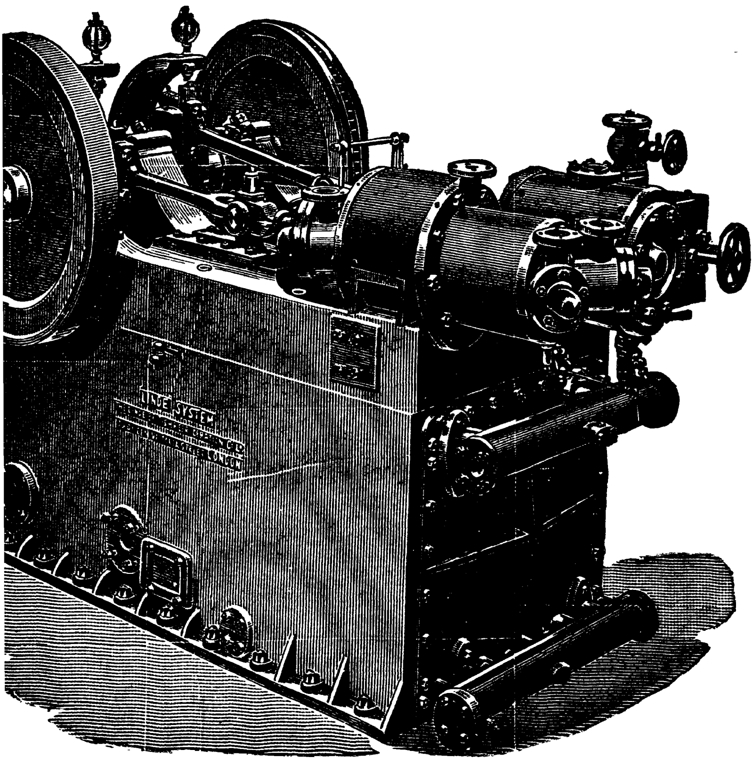
The Linde System of Refrigeration is identical in principle, and only differs in mechanical details from the De La Vergne previously described. It is, therefore, based on the evaporation of liquid anhydrous ammonia and the subsequent liquefaction thereof by means of mechanical compression together with the cooling of the vapour thus formed, so as to enable it to be used over and over again. As will be seen from the accompanying illustration, the self-contained motive-power plant, as chiefly used on board ship, consists of a horizontal steam engine on the right, with a horizontal duplex compressor pump to the left, and an ammonia condenser in the sole plate. As far as the compressor is concerned, the chief differences between the Linde and the De La Vergne systems are:—

(1) That in the former a horizontal compressor is used instead of a vertical one in the latter case.

(2) That a special oil (not susceptible to change at any temperature attainable by the machine, which does not contain any acid or other deleterious matter, and which does not saponify when brought into contact with ammonia is used solely for lubricating purposes. Whereas, the oil used in the De La

Vergne system serves not only as a lubricant to the working parts, but also to partly carry away the heat of compression, and, further, to fill up the clearance spaces, as well as to seal the valves, glands, &c., so as to prevent the escape and consequent inconvenient smell of ammonia.

(3) In the Linde system a small quantity of liquid ammonia is introduced into the compressor during each suction stroke for



Ship Refrigerating Machinery on the Linde System.
By The Linde British Refrigerating Company.

the purpose of cooling the vapour of ammonia. This liquid ammonia evaporates during compression, and thus the heat due to compression, which would otherwise appear as sensible heat, is thereby absorbed and rendered latent in producing the change in the physical state of the liquid. The curve of compression is thus kept down as nearly as may be to the isothermal line, and

the power required for compression is to this extent correspondingly reduced.

Apparatus for Transmitting the Cold Produced to the Chambers requiring Refrigeration.—The following general principles are adopted for transferring the cold generated by the refrigerating machinery to the chambers or rooms requiring to be cooled.

First Method.—An uncongealable solution of salt (chloride of sodium) in water is reduced by the refrigerating machine to a low temperature, and this liquor acts as transmitter of cold in one of the following methods:—

(*a*) The cold brine is constantly circulated from the brine refrigerator through pipes placed in the refrigerated chambers, and returned to the brine cooler. The result is that not only is heat abstracted from the air of the refrigerated rooms, but also a large degree of the moisture which may be present in them. This moisture is condensed on the exterior of the brine pipe systems either in the form of condensed water or hoar-frost. Suitable drip trays are provided, in order to prevent this moisture from falling upon the contents of the rooms. The circulation of air with this system is a moderate one, being produced merely by the differences between the temperatures prevailing near the brine-pipes and those in the lower parts of the rooms.

(*b*) The brine is cooled in a shallow rectangular open tank containing the evaporator coils. On the tank is mounted a number of slowly revolving transverse shafts, and on each shaft is fixed a number of parallel discs, partly immersed in the brine, the entire apparatus being placed in an insulated passage through which an air current is continually passed by a fan, in a direction parallel to the revolving discs. It will be seen, that as the discs revolve and are kept covered by a film of the refrigerated brine, the air passing between the disc-spaces becomes cooled, and produces a low temperature in any chamber or room into which it may be conducted through properly arranged air-trunks. As a rule the air is always taken back from the cold rooms, passed over the discs and returned to the cold rooms, and any required amount of fresh air is introduced by means of adjustable openings in the air-trunks, communicating with the outer atmosphere. In this instance, also, moisture may be removed from the refrigerated rooms and deposited in the brine contained in the trough. No accumulation of frost can take place, and the refrigerated surfaces are always perfectly active. The circumstance of all moisture being deposited in the brine necessitates either a periodical loss of the same or its re-concentration. The fan produces a very effective air circulation within the rooms to

be cooled. This in most cases is extremely desirable, and, as will be readily understood, produces the most beneficial results.

Second Method.—Instead of using an uncongealable liquid as a bearer of cold, the refrigerator coils (in which the vaporisation of the liquid anhydrous ammonia takes place) are sometimes constructed with extra large surfaces, and placed either in the upper part of the rooms to be cooled, or in a separate chamber. In the latter case, a fan constantly circulates the air between this chamber and the refrigerated rooms. This is the system generally adopted on board ships, and has been found to be in all respects most satisfactory. In cases where the air temperature is not sufficiently high to cause a complete removal of the snow deposited on the ammonia-coils, the snow is thawed by the ammonia vapours themselves, the evaporator-coils being for the time used as a condenser. Occasionally the snow is thawed by a current of hot air taken from the outside.

Although all of these methods have been applied on an extensive scale, the system most strongly recommended in cases where its application is possible is the combination of revolving discs immersed in brine. There are no brine or ammonia pipes in the rooms; whilst the rapid air circulation by the fan is easily managed, and has been found in most cases to be requisite for obtaining a satisfactory result as to purity, dryness, and equable temperature in all the rooms.

Where circumstances require the refrigerated rooms to be at a distance from the refrigerating machine, it is generally most convenient to place bundles of brine pipes in each room; but even in such a case, in the event of a small amount of motive power being available close to such rooms, the system of revolving discs and fans can be readily applied, the brine being cooled in a refrigerator near the compressor, and conveyed to and from the disc tanks through insulated pipes.

A large beef-chilling plant on the Linde system was erected in the beginning of 1890 at the Woodside lairage of the Mersey Dock and Harbour Board. It is capable of chilling 660 carcases of beef, each weighing about 9 cwts., from 90 degrees Fahr. to 33 degrees in 17 hours. It consists of a horizontal compound tandem jet-condensing steam engine, which drives a double-acting Linde compressor at the rate of about 65 revolutions per minute, when supplied with steam at 120 lbs. pressure from a marine type boiler. The air-cooling apparatus consists of two disc tanks, placed above the chill rooms, at one end. Each disc system has its own fan, which draws the air from the top of each of the chill rooms, passes it over the discs, and drives it into the rooms at the opposite end to that from which it is withdrawn.

The ammonia condenser is placed in the compressor-room, and is supplied with cooling water by a pump which takes its supply from a well, fed with the drainage water from the Mersey Tunnel. After passing through the ammonia condenser the water is used in the condenser of the steam engine. There are six chill rooms, each about 55 ft. long by 14 ft. wide, and about 13 ft. high. The walls of the rooms are built of brick, with air spaces. The floors are cement, and the ceilings are timber, covered with a layer of fine ashes. The air-cooling apparatus is contained in an insulated casing, which is so arranged as to cause the air to come in contact with the cooled surfaces of the discs.

APPENDIX.

EXAMINATION TABLES.

USEFUL CONSTANTS.

1 Inch = 25·4 millimetres.

1 Gallon = ·1605 cubic foot = 10 lbs. of water at 62° F. ∴ 1 lb. = ·01605 cubic foot.

1 Knot = 6080 feet per hour. 1 Naut = 6080 feet.

Weight of 1 lb. in London = 445,000 dynes.

One pound avoirdupois = 7000 grains = 453·6 grammes.

1 Cubic foot of water weighs 62·3 lbs.

1 Cubic foot of air at 0° C. and 1 atmosphere, weighs ·0807 lb.

1 Cubic foot of Hydrogen at 0° C. and 1 atmosphere, weighs ·00557 lb.

1 Foot-pound = $1·3562 \times 10^7$ ergs.

1 Horse-power-hour = 33000 × 60 foot-pounds.

1 Electrical unit = 1000 watt-hours.

Joule's Equivalent to suit Regnault's H, is $\begin{cases} \text{774 ft.-lbs.} = \text{1 Fah. unit.} \\ \text{1393 ft.-lbs.} = \text{1 Cent. ,,} \end{cases}$

1 Horse-power = 33000 foot-pounds per minute = 746 watts.

Volts × amperes = watts.

1 Atmosphere = 14·7 lb. per square inch = 2116 lbs. per square foot = 760 m.m. of mercury = 10^6 dynes per sq. cm. nearly.

A Column of water 2·3 feet high corresponds to a pressure of 1 lb. per square inch.

Absolute temp., $t = \theta°$ C. + 273°·7.

Regnault's H = 606·5 + ·305 θ° C. = 1082 + ·305 θ° F.

$$p\,u^{1.0646} = 479$$

$$\log_{10} p = 6·1007 - \frac{B}{t} - \frac{C}{t^2}$$

where $\log_{10} B = 3·1812$, $\log_{10} C = 5·0871$,

p is in pounds per square inch, t is absolute temperature Centigrade, u is the volume in cubic feet per pound of steam.

$$\pi = 3·1416 = \frac{22}{7} = \frac{355}{113} = 10(\sqrt{3} - \sqrt{2}).$$

One radian = 57·3 degrees.

To convert common into Napierian logarithms, multiply by 2·3026.

The base of the Napierian logarithm is e = 2·7183.

The value of g at London = 32·182 feet per second per second.

TABLE OF LOGARITHMS.

	0	1	2	3	4	5	6	7	8	9	1	2	3	4	5	6	7	8	9
10	0000	0043	0086	0128	0170	0212	0253	0294	0334	0374	4	8	12	17	21	25	29	33	37
11	0414	0453	0492	0531	0569	0607	0645	0682	0719	0755	4	8	11	15	19	23	26	30	34
12	0792	0828	0864	0899	0934	0969	1004	1038	1072	1106	3	7	10	14	17	21	24	28	31
13	1139	1173	1206	1239	1271	1303	1335	1367	1399	1430	3	6	10	13	16	19	23	26	29
14	1461	1492	1523	1553	1584	1614	1644	1673	1703	1732	3	6	9	12	15	18	21	24	27
15	1761	1790	1818	1847	1875	1903	1931	1959	1987	2014	3	6	8	11	14	17	20	22	25
16	2041	2068	2095	2122	2148	2175	2201	2227	2253	2279	3	5	8	11	13	16	18	21	24
17	2304	2330	2355	2380	2405	2430	2455	2480	2504	2529	2	5	7	10	12	15	17	20	22
18	2553	2577	2601	2625	2648	2672	2695	2718	2742	2765	2	5	7	9	12	14	16	19	21
19	2788	2810	2833	2856	2878	2900	2923	2945	2967	2989	2	4	7	9	11	13	16	18	20
20	3010	3032	3054	3075	3096	3118	3139	3160	3181	3201	2	4	6	8	11	13	15	17	19
21	3222	3243	3263	3284	3304	3324	3345	3365	3385	3404	2	4	6	8	10	12	14	16	18
22	3424	3444	3464	3483	3502	3522	3541	3560	3579	3598	2	4	6	8	10	12	14	15	17
23	3617	3636	3655	3674	3692	3711	3729	3747	3766	3784	2	4	6	7	9	11	13	15	17
24	3802	3820	3838	3856	3874	3892	3909	3927	3945	3962	2	4	5	7	9	11	12	14	16
25	3979	3997	4014	4031	4048	4065	4082	4099	4116	4133	2	3	5	7	9	10	12	14	15
26	4150	4166	4183	4200	4216	4232	4249	4265	4281	4298	2	3	5	7	8	10	11	13	15
27	4314	4330	4346	4362	4378	4393	4409	4425	4440	4456	2	3	5	6	8	9	11	13	14
28	4472	4487	4502	4518	4533	4548	4564	4579	4594	4609	2	3	5	6	8	9	11	12	14
29	4624	4639	4654	4669	4683	4698	4713	4728	4742	4757	1	3	4	6	7	9	10	12	13
30	4771	4786	4800	4814	4829	4843	4857	4871	4886	4900	1	3	4	6	7	9	10	11	13
31	4914	4928	4942	4955	4969	4983	4997	5011	5024	5038	1	3	4	6	7	8	10	11	12
32	5051	5065	5079	5092	5105	5119	5132	5145	5159	5172	1	3	4	5	7	8	9	11	12
33	5185	5198	5211	5224	5237	5250	5263	5276	5289	5302	1	3	4	5	6	8	9	10	12
34	5315	5328	5340	5353	5366	5378	5391	5403	5416	5428	1	3	4	5	6	8	9	10	11
35	5441	5453	5465	5478	5490	5502	5514	5527	5539	5551	1	2	4	5	6	7	9	10	11
36	5563	5575	5587	5599	5611	5623	5635	5647	5658	5670	1	2	4	5	6	7	8	10	11
37	5682	5694	5705	5717	5729	5740	5752	5763	5775	5786	1	2	3	5	6	7	8	9	10
38	5798	5809	5821	5832	5843	5855	5866	5877	5888	5899	1	2	3	5	6	7	8	9	10
39	5911	5922	5933	5944	5955	5966	5977	5988	5999	6010	1	2	3	4	5	7	8	9	10
40	6021	6031	6042	6053	6064	6075	6085	6096	6107	6117	1	2	3	4	5	6	8	9	10
41	6128	6138	6149	6160	6170	6180	6191	6201	6212	6222	1	2	3	4	5	6	7	8	9
42	6232	6243	6253	6263	6274	6284	6294	6304	6314	6325	1	2	3	4	5	6	7	8	9
43	6335	6345	6355	6365	6375	6385	6395	6405	6415	6425	1	2	3	4	5	6	7	8	9
44	6435	6444	6454	6464	6474	6484	6493	6503	6513	6522	1	2	3	4	5	6	7	8	9
45	6532	6542	6551	6561	6571	6580	6590	6599	6609	6618	1	2	3	4	5	6	7	8	9
46	6628	6637	6646	6656	6665	6675	6684	6693	6702	6712	1	2	3	4	5	6	7	7	8
47	6721	6730	6739	6749	6758	6767	6776	6785	6794	6803	1	2	3	4	5	5	6	7	8
48	6812	6821	6830	6839	6848	6857	6866	6875	6884	6893	1	2	3	4	4	5	6	7	8
49	6902	6911	6920	6928	6937	6946	6955	6964	6972	6981	1	2	3	4	4	5	6	7	8
50	6990	6998	7007	7016	7024	7033	7042	7050	7059	7067	1	2	3	3	4	5	6	7	8
51	7076	7084	7093	7101	7110	7118	7126	7135	7143	7152	1	2	3	3	4	5	6	7	8
52	7160	7168	7177	7185	7193	7202	7210	7218	7226	7235	1	2	2	3	4	5	6	7	7
53	7243	7251	7259	7267	7275	7284	7292	7300	7308	7316	1	2	2	3	4	5	6	6	7
54	7324	7332	7340	7348	7356	7364	7372	7380	7388	7396	1	2	2	3	4	5	6	6	7

TABLE OF LOGARITHMS.—*Continued.*

	0	1	2	3	4	5	6	7	8	9	1	2	3	4	5	6	7	8	9
55	7404	7412	7419	7427	7435	7443	7451	7459	7466	7474	1	2	2	3	4	5	5	6	7
56	7482	7490	7497	7505	7513	7520	7528	7536	7543	7551	1	2	2	3	4	5	5	6	7
57	7559	7566	7574	7582	7589	7597	7604	7612	7619	7627	1	2	2	3	4	5	5	6	7
58	7634	7642	7649	7657	7664	7672	7679	7686	7694	7701	1	1	2	3	4	4	5	6	7
59	7709	7716	7723	7731	7738	7745	7752	7760	7767	7774	1	1	2	3	4	4	5	6	7
60	7782	7789	7796	7803	7810	7818	7825	7832	7839	7846	1	1	2	3	4	4	5	6	6
61	7853	7860	7868	7875	7882	7889	7896	7903	7910	7917	1	1	2	3	4	4	5	6	6
62	7924	7931	7938	7945	7952	7959	7966	7973	7980	7987	1	1	2	3	3	4	5	6	6
63	7993	8000	8007	8014	8021	8028	8035	8041	8048	8055	1	1	2	3	3	4	5	5	6
64	8062	8069	8075	8082	8089	8096	8102	8109	8116	8122	1	1	2	3	3	4	5	5	6
65	8129	8136	8142	8149	8156	8162	8169	8176	8182	8189	1	1	2	3	3	4	5	5	6
66	8195	8202	8209	8215	8222	8228	8235	8241	8248	8254	1	1	2	3	3	4	5	5	6
67	8261	8267	8274	8280	8287	8293	8299	8306	8312	8319	1	1	2	3	3	4	5	5	6
68	8325	8331	8338	8344	8351	8357	8363	8370	8376	8382	1	1	2	3	3	4	4	5	6
69	8388	8395	8401	8407	8414	8420	8426	8432	8439	8445	1	1	2	2	3	4	4	5	6
70	8451	8457	8463	8470	8476	8482	8488	8494	8500	8506	1	1	2	2	3	4	4	5	6
71	8513	8519	8525	8531	8537	8543	8549	8555	8561	8567	1	1	2	2	3	4	4	5	5
72	8573	8579	8585	8591	8597	8603	8609	8615	8621	8627	1	1	2	2	3	4	4	5	5
73	8633	8639	8645	8651	8657	8663	8669	8675	8681	8686	1	1	2	2	3	4	4	5	5
74	8692	8698	8704	8710	8716	8722	8727	8733	8739	8745	1	1	2	2	3	4	4	5	5
75	8751	8756	8762	8768	8774	8779	8785	8791	8797	8802	1	1	2	2	3	3	4	5	5
76	8808	8814	8820	8825	8831	8837	8842	8848	8854	8859	1	1	2	2	3	3	4	5	5
77	8865	8871	8876	8882	8887	8893	8899	8904	8910	8915	1	1	2	2	3	3	4	4	5
78	8921	8927	8932	8938	8943	8949	8954	8960	8965	8971	1	1	2	2	3	3	4	4	5
79	8976	8982	8987	8993	8998	9004	9009	9015	9020	9025	1	1	2	2	3	3	4	4	5
80	9031	9036	9042	9047	9053	9058	9063	9069	9074	9079	1	1	2	2	3	3	4	4	5
81	9085	9090	9096	9101	9106	9112	9117	9122	9128	9133	1	1	2	2	3	3	4	4	5
82	9138	9143	9149	9154	9159	9165	9170	9175	9180	9186	1	1	2	2	3	3	4	4	5
83	9191	9196	9201	9206	9212	9217	9222	9227	9232	9238	1	1	2	2	3	3	4	4	5
84	9243	9248	9253	9258	9263	9269	9274	9279	9284	9289	1	1	2	2	3	3	4	4	5
85	9294	9299	9304	9309	9315	9320	9325	9330	9335	9340	1	1	2	2	3	3	4	4	5
86	9345	9350	9355	9360	9365	9370	9375	9380	9385	9390	1	1	2	2	3	3	4	4	5
87	9395	9400	9405	9410	9415	9420	9425	9430	9435	9440	0	1	1	2	2	3	3	4	4
88	9445	9450	9455	9460	9465	9469	9474	9479	9484	9489	0	1	1	2	2	3	3	4	4
89	9494	9499	9504	9509	9513	9518	9523	9528	9533	9538	0	1	1	2	2	3	3	4	4
90	9542	9547	9552	9557	9562	9566	9571	9576	9581	9586	0	1	1	2	2	3	3	4	4
91	9590	9595	9600	9605	9609	9614	9619	9624	9628	9633	0	1	1	2	2	3	3	4	4
92	9638	9643	9647	9652	9657	9661	9666	9671	9675	9680	0	1	1	2	2	3	3	4	4
93	9685	9689	9694	9699	9703	9708	9713	9717	9722	9727	0	1	1	2	2	3	3	4	4
94	9731	9736	9741	9745	9750	9754	9759	9763	9768	9773	0	1	1	2	2	3	3	4	4
95	9777	9782	9786	9791	9795	9800	9805	9809	9814	9818	0	1	1	2	2	3	3	4	4
96	9823	9827	9832	9836	9841	9845	9850	9854	9859	9863	0	1	1	2	2	3	3	4	4
97	9868	9872	9877	9881	9886	9890	9894	9899	9903	9908	0	1	1	2	2	3	3	4	4
98	9912	9917	9921	9926	9930	9934	9939	9943	9948	9952	0	1	1	2	2	3	3	4	4
99	9956	9961	9965	9969	9974	9978	9983	9987	9991	9996	0	1	1	2	2	3	3	3	4

TABLE OF ANTILOGARITHMS.

	0	1	2	3	4	5	6	7	8	9	1	2	3	4	5	6	7	8	9
·00	1000	1002	1005	1007	1009	1012	1014	1016	1019	1021	0	0	1	1	1	1	2	2	2
·01	1023	1026	1028	1030	1033	1035	1038	1040	1042	1045	0	0	1	1	1	1	2	2	2
·02	1047	1050	1052	1054	1057	1059	1062	1064	1067	1069	0	0	1	1	1	1	2	2	2
·03	1072	1074	1076	1079	1081	1084	1086	1089	1091	1094	0	0	1	1	1	1	2	2	2
·04	1096	1099	1102	1104	1107	1109	1112	1114	1117	1119	0	1	1	1	1	2	2	2	2
·05	1122	1125	1127	1130	1132	1135	1138	1140	1143	1146	0	1	1	1	1	2	2	2	2
·06	1148	1151	1153	1156	1159	1161	1164	1167	1169	1172	0	1	1	1	1	2	2	2	2
·07	1175	1178	1180	1183	1186	1189	1191	1194	1197	1199	0	1	1	1	1	2	2	2	2
·08	1202	1205	1208	1211	1213	1216	1219	1222	1225	1227	0	1	1	1	1	2	2	2	3
·09	1230	1233	1236	1239	1242	1245	1247	1250	1253	1256	0	1	1	1	1	2	2	2	3
·10	1259	1262	1265	1268	1271	1274	1276	1279	1282	1285	0	1	1	1	1	2	2	2	3
·11	1288	1291	1294	1297	1300	1303	1306	1309	1312	1315	0	1	1	1	2	2	2	2	3
·12	1318	1321	1324	1327	1330	1334	1337	1340	1343	1346	0	1	1	1	2	2	2	2	3
·13	1349	1352	1355	1358	1361	1365	1368	1371	1374	1377	0	1	1	1	2	2	2	3	3
·14	1380	1384	1387	1390	1393	1396	1400	1403	1406	1409	0	1	1	1	2	2	2	3	3
·15	1413	1416	1419	1422	1426	1429	1432	1435	1439	1442	0	1	1	1	2	2	2	3	3
·16	1445	1449	1452	1455	1459	1462	1466	1469	1472	1476	0	1	1	1	2	2	2	3	3
·17	1479	1483	1486	1489	1493	1496	1500	1503	1507	1510	0	1	1	1	2	2	2	3	3
·18	1514	1517	1521	1524	1528	1531	1535	1538	1542	1545	0	1	1	1	2	2	2	3	3
·19	1549	1552	1556	1560	1563	1567	1570	1574	1578	1581	0	1	1	1	2	2	3	3	3
·20	1585	1589	1592	1596	1600	1603	1607	1611	1614	1618	0	1	1	1	2	2	3	3	3
·21	1622	1626	1629	1633	1637	1641	1644	1648	1652	1656	0	1	1	2	2	2	3	3	3
·22	1660	1663	1667	1671	1675	1679	1683	1687	1690	1694	0	1	1	2	2	2	3	3	3
·23	1698	1702	1706	1710	1714	1718	1722	1726	1730	1734	0	1	1	2	2	2	3	3	4
·24	1738	1742	1746	1750	1754	1758	1762	1766	1770	1774	0	1	1	2	2	2	3	3	4
·25	1778	1782	1786	1791	1795	1799	1803	1807	1811	1816	0	1	1	2	2	2	3	3	4
·26	1820	1824	1828	1832	1837	1841	1845	1849	1854	1858	0	1	1	2	2	3	3	3	4
·27	1862	1866	1871	1875	1879	1884	1888	1892	1897	1901	0	1	1	2	2	3	3	3	4
·28	1905	1910	1914	1919	1923	1928	1932	1936	1941	1945	0	1	1	2	2	3	3	4	4
·29	1950	1954	1959	1963	1968	1972	1977	1982	1986	1991	0	1	1	2	2	3	3	4	4
·30	1995	2000	2004	2009	2014	2018	2023	2028	2032	2037	0	1	1	2	2	3	3	4	4
·31	2042	2046	2051	2056	2061	2065	2070	2075	2080	2084	0	1	1	2	2	3	3	4	4
·32	2089	2094	2099	2104	2109	2113	2118	2123	2128	2133	0	1	1	2	2	3	3	4	4
·33	2138	2143	2148	2153	2158	2163	2168	2173	2178	2183	0	1	1	2	2	3	3	4	4
·34	2188	2193	2198	2203	2208	2213	2218	2223	2228	2234	1	1	2	2	3	3	4	4	5
·35	2239	2244	2249	2254	2259	2265	2270	2275	2280	2286	1	1	2	2	3	3	4	4	5
·36	2291	2296	2301	2307	2312	2317	2323	2328	2333	2339	1	1	2	2	3	3	4	4	5
·37	2344	2350	2355	2360	2366	2371	2377	2382	2388	2393	1	1	2	2	3	3	4	4	5
·38	2399	2404	2410	2415	2421	2427	2432	2438	2443	2449	1	1	2	2	3	3	4	4	5
·39	2455	2460	2466	2472	2477	2483	2489	2495	2500	2506	1	1	2	2	3	3	4	5	5
·40	2512	2518	2523	2529	2535	2541	2547	2553	2559	2564	1	1	2	2	3	4	4	5	5
·41	2570	2576	2582	2588	2594	2600	2606	2612	2618	2624	1	1	2	2	3	4	4	5	5
·42	2630	2636	2642	2649	2655	2661	2667	2673	2679	2685	1	1	2	2	3	4	4	5	6
·43	2692	2698	2704	2710	2716	2723	2729	2735	2742	2748	1	1	2	3	3	4	4	5	6
·44	2754	2761	2767	2773	2780	2786	2793	2799	2805	2812	1	1	2	3	3	4	4	5	6
·45	2818	2825	2831	2838	2844	2851	2858	2864	2871	2877	1	1	2	3	3	4	5	5	6
·46	2884	2891	2897	2904	2911	2917	2924	2931	2938	2944	1	1	2	3	3	4	5	5	6
·47	2951	2958	2965	2972	2979	2985	2992	2999	3006	3013	1	1	2	3	3	4	5	5	6
·48	3020	3027	3034	3041	3048	3055	3062	3069	3076	3083	1	1	2	3	4	4	5	6	6
·49	3090	3097	3105	3112	3119	3126	3133	3141	3148	3155	1	1	2	3	4	4	5	6	6

TABLE OF ANTILOGARITHMS.—*Continued.*

	0	1	2	3	4	5	6	7	8	9	1	2	3	4	5	6	7	8	9
·50	3162	3170	3177	3184	3192	3199	3206	3214	3221	3228	1	1	2	3	4	4	5	6	7
·51	3236	3243	3251	3258	3266	3273	3281	3289	3296	3304	1	2	2	3	4	5	5	6	7
·52	3311	3319	3327	3334	3342	3350	3357	3365	3373	3381	1	2	2	3	4	5	5	6	7
·53	3388	3396	3404	3412	3420	3428	3436	3443	3451	3459	1	2	2	3	4	5	6	6	7
·54	3467	3475	3483	3491	3499	3508	3516	3524	3532	3540	1	2	2	3	4	5	6	6	7
·55	3548	3556	3565	3573	3581	3589	3597	3606	3614	3622	1	2	2	3	4	5	6	7	7
·56	3631	3639	3648	3656	3664	3673	3681	3690	3698	3707	1	2	3	3	4	5	6	7	8
·57	3715	3724	3733	3741	3750	3758	3767	3776	3784	3793	1	2	3	3	4	5	6	7	8
·58	3802	3811	3819	3828	3837	3846	3855	3864	3873	3882	1	2	3	4	4	5	6	7	8
·59	3890	3899	3908	3917	3926	3936	3945	3954	3963	3972	1	2	3	4	5	5	6	7	8
·60	3981	3990	3999	4009	4018	4027	4036	4046	4055	4064	1	2	3	4	5	6	6	7	8
·61	4074	4083	4093	4102	4111	4121	4130	4140	4150	4159	1	2	3	4	5	6	7	8	9
·62	4169	4178	4188	4198	4207	4217	4227	4236	4246	4256	1	2	3	4	5	6	7	8	9
·63	4266	4276	4285	4295	4305	4315	4325	4335	4345	4355	1	2	3	4	5	6	7	8	9
·64	4365	4375	4385	4395	4406	4416	4426	4436	4446	4457	1	2	3	4	5	6	7	8	9
·65	4467	4477	4487	4498	4508	4519	4529	4539	4550	4560	1	2	3	4	5	6	7	8	9
·66	4571	4581	4592	4603	4613	4624	4634	4645	4656	4567	1	2	3	4	5	6	7	9	10
·67	4677	4688	4699	4710	4721	4732	4742	4753	4764	4775	1	2	3	4	5	7	8	9	10
·68	4786	4797	4808	4819	4831	4842	4853	4864	4875	4887	1	2	3	4	6	7	8	9	10
·69	4898	4909	4920	4932	4943	4955	4966	4977	4989	5000	1	2	3	5	6	7	8	9	10
·70	5012	5023	5035	5047	5058	5070	5082	5093	5105	5117	1	2	4	5	6	7	8	9	11
·71	5129	5140	5152	5164	5176	5188	5200	5212	5224	5236	1	2	4	5	6	7	8	10	11
·72	5248	5260	5272	5284	5297	5309	5321	5333	5346	5358	1	2	4	5	6	7	9	10	11
·73	5370	5383	5395	5408	5420	5433	5445	5458	5470	5483	1	3	4	5	6	8	9	10	11
·74	5495	5508	5521	5534	5546	5559	5572	5585	5598	5610	1	3	4	5	6	8	9	10	12
·75	5623	5636	5649	5662	5675	5689	5702	5715	5728	5741	1	3	4	5	7	8	9	10	12
·76	5754	5768	5781	5794	5808	5821	5834	5848	5861	5875	1	3	4	5	7	8	9	11	12
·77	5888	5902	5916	5929	5943	5957	5970	5984	5998	6012	1	3	4	5	7	8	10	11	12
·78	6026	6039	6053	6067	6081	6095	6109	6124	6138	6152	1	3	4	6	7	8	10	11	13
·79	6166	6180	6194	6209	6223	6237	6252	6266	6281	6295	1	3	4	6	7	9	10	11	13
·80	6310	6324	6339	6353	6368	6383	6397	6412	6427	6442	1	3	4	6	7	9	10	12	13
·81	6457	6471	6486	6501	6516	6531	6546	6561	6577	6592	2	3	5	6	8	9	11	12	14
·82	6607	6622	6637	6653	6668	6683	6699	6714	6730	6745	2	3	5	6	8	9	11	12	14
·83	6761	6776	6792	6808	6823	6839	6855	6871	6887	6902	2	3	5	6	8	9	11	13	14
·84	6918	6934	6950	6966	6982	6998	7015	7031	7047	7063	2	3	5	6	8	10	11	13	15
·85	7079	7096	7112	7129	7145	7161	7178	7194	7211	7228	2	3	5	7	8	10	12	13	15
·86	7244	7261	7278	7295	7311	7328	7345	7362	7379	7396	2	3	5	7	8	10	12	13	15
·87	7413	7430	7447	7464	7482	7499	7516	7534	7551	7568	2	3	5	7	9	10	12	14	16
·88	7586	7603	7621	7638	7656	7674	7691	7709	7727	7745	2	4	5	7	9	11	12	14	16
·89	7762	7780	7798	7816	7834	7852	7870	7889	7907	7925	2	4	5	7	9	11	13	14	16
·90	7943	7962	7980	7998	8017	8035	8054	8072	8891	8110	2	4	6	7	9	11	13	15	17
·91	8128	8147	8166	8185	8204	8222	8241	8260	8279	8299	2	4	6	8	9	11	13	15	17
·92	8318	8337	8356	8375	8395	8414	8433	8453	8472	8492	2	4	6	8	10	12	14	15	17
·93	8511	8531	8551	8570	8590	8610	8630	8650	8670	8690	2	4	6	8	10	12	14	16	18
·94	8710	8730	8750	8770	8790	8810	8831	8851	8872	8892	2	4	6	8	10	12	14	16	18
·95	8913	8933	8954	8974	8995	9016	9036	9057	9078	9099	2	4	6	8	10	12	15	17	19
·96	9120	9141	9162	9183	9204	9226	9247	9268	9290	9311	2	4	6	8	11	13	15	17	19
·97	9333	9354	9376	9397	9419	9441	9462	9484	9506	9528	2	4	7	9	11	13	15	17	20
·98	9550	9572	9594	9616	9638	9661	9683	9705	9727	9750	2	4	7	9	11	13	16	18	20
·99	9772	9795	9817	9840	9863	9886	9908	9931	9954	9977	2	5	7	9	11	14	16	18	20

TABLE OF FUNCTIONS OF ANGLES.

Angle.		Chord.	Sine.	Tangent.	Co-tangent.	Cosine.			
Degrees.	Radians.								
0°	0	000	0	0	∞	1	1·414	1·5708	90°
1	·0175	·017	·0175	·0175	57·2900	·9998	1·402	1·5533	89
2	·0349	·035	·0349	·0349	28·6363	·9994	1·389	1·5359	88
3	·0524	·052	·0523	·0524	19·0811	·9986	1·377	1·5184	87
4	·0698	·070	·0698	·0699	14·3007	·9976	1·364	1·5010	86
5	·0873	·087	·0872	·0875	11·4301	·9962	1·351	1·4835	85
6	·1047	·105	·1045	·1051	9·5144	·9945	1·338	1·4661	84
7	·1222	·122	·1219	·1228	8·1443	·9925	1·325	1·4486	83
8	·1396	·140	·1392	·1405	7·1154	·9903	1·312	1·4312	82
9	·1571	·157	·1564	·1584	6·3138	·9877	1·299	1·4137	81
10	·1745	·174	·1736	·1763	5·6713	·9848	1·286	1·3963	80
11	·1929	·192	·1908	·1944	5·1446	·9816	1·272	1·3788	79
12	·2094	·209	·2079	·2126	4·7046	·9781	1·259	1·3614	78
13	·2269	·226	·2250	·2309	4·3315	·9744	1·245	1·3439	77
14	·2443	·244	·2419	·2493	4·0108	·9703	1·231	1·3265	76
15	·2618	·261	·2588	·2679	3·7321	·9659	1·218	1·3090	75
16	·2793	·278	·2756	·2867	3·4874	·9613	1·204	1·2915	74
17	·2967	·296	·2924	·3057	3·2709	·9563	1·190	1·2741	73
18	·3142	·313	·3090	·3249	3·0777	·9511	1·176	1·2566	72
19	·3316	·330	·3256	·3443	2·9042	·9455	1·161	1·2392	71
20	·3491	·347	·3420	·3640	2·7475	·9397	1·147	1·2217	70
21	·3665	·364	·3584	·3839	2·6051	·9336	1·133	1·2043	69
22	·3840	·382	·3746	·4040	2·4751	·9272	1·118	1·1868	68
23	·4014	·399	·3907	·4245	2·3559	·9205	1·104	1·1694	67
24	·4189	·416	·4067	·4452	2·2460	·9135	1·089	1·1519	66
25	·4363	·433	·4226	·4663	2·1445	·9063	1·075	1·1345	65
26	·4538	·450	·4384	·4877	2·0503	·8988	1·060	1·1170	64
27	·4712	·467	·4540	·5095	1·9626	·8910	1·045	1·0996	63
28	·4887	·484	·4695	·5317	1·8807	·8829	1·030	1·0821	62
29	·5061	·501	·4848	·5543	1·8040	·8746	1·015	1·0647	61
30	·5236	·518	·5000	·5774	1·7321	·8660	1·000	1·0472	60
31	·5411	·534	·5150	·6009	1·6643	·8572	·985	1·0297	59
32	·5585	·551	·5299	·6249	1·6003	·8480	·970	1·0123	58
33	·5760	·568	·5446	·6494	1·5399	·8387	·954	·9948	57
34	·5934	·585	·5592	·6745	1·4826	·8290	·939	·9774	56
35	·6109	·601	·5736	·7002	1·4281	·8192	·923	·9599	55
36	·6283	·618	·5878	·7265	1·3764	·8090	·908	·9425	54
37	·6458	·635	·6018	·7536	1·3270	·7986	·892	·9250	53
38	·6632	·651	·6157	·7813	1·2799	·7880	·877	·9076	52
39	·6807	·668	·6293	·8098	1·2349	·7771	·861	·8901	51
40	·6981	·684	·6428	·8391	1·1918	·7660	·845	·8727	50
41	·7156	·700	·6561	·8693	1·1504	·7547	·829	·8552	49
42	·7330	·717	·6691	·9004	1·1106	·7431	·813	·8378	48
43	·7505	·733	·6820	·9325	1·0724	·7314	·797	·8203	47
44	·7679	·749	·6947	·9657	1·0355	·7193	·781	·8029	46
45°	·7854	·765	·7071	1·0000	1·0000	·7071	·765	·7854	45°
			Cosine.	Co-tangent.	Tangent.	Sine.	Chord.	Radians.	Degrees.
								Angle.	

The Institution of Civil Engineers.

EXTRACTS FROM RULES AND SYLLABUS OF EXAMINATIONS FOR

ELECTION OF ASSOCIATE MEMBERS.

PART II.*—*Scientific Knowledge.*

SECTION A.

1. **Mechanics** (one Paper, *time allowed,* 3 *hours*).
2. **Strength and Elasticity of Materials** (one Paper, *time allowed,* 3 *hours*).
3. *Either* (*a*) **Theory of Structures,**
   or (*b*) **Theory of Electricity and Magnetism** (one Paper, *time allowed,* 3 *hours*).

SECTION B

Two of the following nine subjects—not more than one from any group (one Paper in each subject taken, *time allowed,* 3 *hours for each Paper*):—

Group i.	*Group* ii.	*Group* iii.
Geodesy.	Hydraulics.	Geology and Mineralogy.
Theory of Heat Engines.	Theory of Machines.	Stability and Resistance of Ships.
Thermo- and Electro-Chemistry.	Metallurgy.	Applications of Electricity.

Mathematics.—The standard of Mathematics required for the Papers in Part II. of the examination is that of the mathematical portion of the Examination for the Admission of Students, though questions may be set involving the use of higher Mathematics.

* Candidates may offer themselves for examination in Sections A and B of Part II. together; or they may enter for Section A alone, and, if successful, may take Section B at a subsequent examination. In the latter case, however, such candidates will not be allowed to present themselves for examination in Section B unless or until they are actually occupied in work as pupils or assistants to Corporate Engineers. The Council may permit Candidates who have attempted the whole of Part II. at one examination, and have failed in Section B only, to complete their qualification by passing in that section at a subsequent examination, subject to their being then occupied as above stated.

The range of the examinations in the several subjects, in each of which a choice of questions will be allowed, is indicated generally hereunder :—

SECTION A.

1. Mechanics :—

Statics.—Forces acting on a rigid body; moments of forces, composition, and resolution of forces; couples, conditions of equilibrium, with application to loaded structures. The foregoing subjects to be treated both graphically and by aid of algebra and geometry.

Hydrostatics.—Pressure at any point in a gravitating liquid; centre of pressure on immersed plane areas; specific gravity.

Kinematics of Plane Motion.—Velocity and acceleration of a point; instantaneous centre of a moving body.

Kinetics of Plane Motion. — Force, mass, momentum, moment of momentum, work, energy, their relation and their measure; equations of motion of a particle; rectilinear motion under the action of gravity; falling bodies and motion on an inclined plane; motion in a circle; centres of mass and moments of inertia; rotation of a rigid body about a fixed axis; conservation of energy.

2. Strength and Elasticity of Materials :—

Physical properties and elastic constants of cast iron, wrought iron, steel, timber, stone, and cement; relation of stress and strain, limit of elasticity, yield-point, Young's modulus; coefficient of rigidity; extension and lateral contraction; resistance within the elastic limit in tension, compression, shear and torsion; thin shells; strength and deflection in simple cases of bending; beams of uniform resistance; suddenly applied loads.

Ultimate strength with different modes of loading; plasticity, working stress; phenomena in an ordinary tensile test; stress-strain diagram; elongation and contraction of area; effects of hardening, tempering and annealing; fatigue of metals; measurement of hardness.

Forms and arrangements of testing machines for tension, compression, torsion, and bending tests; instruments for measuring extension, compression, and twist; forms of test pieces and arrangements for holding them; influence of form on strength and elongation; methods of ordinary commercial testing; percentage of elongation and contraction of area; test conditions in specifications for cast iron, mild steel, and cement.

3. (*a*) Theory of Structures :—

Graphic and analytic methods for the calculation of bending moments and of shearing forces, and of the stresses in individual members of framework structures loaded at the joints; plate and box girders; incomplete and redundant frames; stresses suddenly applied, and effects of impact; buckling of struts; effect of different end fastenings on their resistance; combined strains; calculations connected with statically indeterminate problems, as beams supported at three points, &c.; travelling loads; riveted and pin-joint girders; rigid and hinged arches; strains due to weight of structures; theory of earth-pressure and of foundations; stability of masonry and brickwork structures.

3. (*b*) Theory of Electricity and Magnetism :—

Electrical and magnetic laws, units, standards, and measurements; electrical and magnetic measuring instruments; the theory of the generation, storage, transformation, and distribution of electrical energy; continuous and alternating currents; arc and incandescent lamps; secondary cells.

SECTION B.

Group i. Theory of Heat Engines :—

Thermodynamic laws; internal and external work; graphical representation of changes in the condition of a fluid; theory of heat engines working with a perfect gas; air- and gas-engine cycles; reversibility, conditions necessary for maximum possible efficiency in any cycle; properties of steam; the Carnot and Clausius cycles; entropy and entropy-temperature diagrams, and their application in the study of heat engines; actual heat engine cycles and their thermodynamic losses; effects of clearance and throttling; initial condensation; testing of heat engines, and the apparatus employed; performances of typical engines of different classes; efficiency.

Group ii. Hydraulics :—

The laws of the flow of water by orifices, notches, and weirs; laws of fluid friction; steady flow in pipes or channels of uniform section; resistance of valves and bends; general phenomena of flow in rivers; methods of determining the discharge of streams; tidal action; generation and effect of waves; impulse and reaction of jets of water; transmission of energy by fluids; principles of machines acting by weight, pressure, and kinetic energy of water; theory and structure of turbines and pumps.

Theory of Machines :—

Kinematics of machines; inversion of kinematic chains; virtual centres; belt, rope, chain, toothed and screw gearing; velocity, acceleration and effort diagrams; inertia of reciprocating parts; elementary cases of balancing; governors and flywheels; friction and efficiency; strength and proportions of machine parts in simple cases.

Group iii. Applications of Electricity :—

Theory and design of continuous- and alternating-current generators and motors, synchronous and induction motors and static transformers; design of generating- and sub-stations and the principal plant required in them; the principal systems of distributing electrical energy, including the arrangement of mains and feeders; estimation of losses and of efficiency; principal systems of electric traction; construction and efficiency of the principal types of electric lamps.

☞ Candidates should see, that all their "Forms" are *duly completed* and *passed* by the Council of the Institution of Civil Engineers, Great George Street, Westminster, S.W., *before* 1st January for the February Examination, and *before* the 1st September for the October Examination.

Examinations Abroad.—The papers of the *October* Examination *only* will be placed before accepted Candidates in India and the Colonies. To enable the Secretary to make arrangements for the Application Forms and Fees, &c., of these Candidates, their Forms, &c., *must* be in the Secretary's hands, *before* the 1st June preceding the October Examinations.

THE INSTITUTION OF CIVIL ENGINEERS' EXAMINATION, FEBRUARY, 1909.

ELECTION OF ASSOCIATE MEMBERS.

Group II.—HYDRAULICS.

Not more than EIGHT *questions to be attempted by any Candidate.*

1. Show how to determine the resultant pressure on a curved surface subjected to a uniform hydrostatic pressure. A 12-inch water main has a bend subtending an angle of 60° at its centre of curvature. The water in the pipe is under a mean head of 65 feet. Determine the magnitude and position of the force tending to displace the bend.

2. An area of 5 square miles forms the catchment basin for a reservoir, and the maximum rainfall is 1·75 inches in 24 hours. In times of flood the water is diverted by a separating weir 36 feet long. Describe the construction of one form of weir suitable for this purpose, and calculate the average height of the water over the weir crest, assuming that the maximum rainfall for one day will pass over the weir in 36 hours, and that the coefficient of the weir is 0·7.

3. Obtain a general expression for the flow through a submerged orifice and apply it to determine the time required to empty a lock, of 15,000 square feet of superficial area and having vertical walls, if the initial difference of level between the lock and the lower reach of the canal is 8 feet, the area of the sluice openings is 30 square feet, and the coefficient of discharge is 0·64. The level of the lower reach of the canal is assumed not to be influenced by the discharge.

4. Describe by aid of sketches the construction of one form of meter suitable for measuring small quantities of water supplied at irregular intervals, such as occurs in a house supply. Explain what are the special features of advantage of the form you describe.

5. Explain what is meant by (1) a free spiral vortex, (2) a forced vortex, and give two examples of each kind of vortex motion which occur in engineering practice. The wheel of a centrifugal pump has an external diameter of 3 feet, and an internal diameter of 1·5 feet. The pump casting is filled with water, and with the discharge pipe closed, the wheel is revolved at the rate of 300 revolutions per minute. Calculate the difference of head set up between the inner and outer rims of the wheel.

6. Explain concisely the special features of difficulty which arise in regulating the speed of a turbine, and describe by aid of sketches the

essential features of one form of governing gear suitable for a turbine of about 1,000 horse-power.

7. Describe the principal features of construction of rectangular measuring weirs ordinarily used for waterworks purposes, and the general conditions to be observed in order to obtain an accurate measure of the flow. Deduce a formula for the flow from such a weir in terms of its dimensions and the head.

8. Explain under what circumstances it is useful to plot the logarithmic values of the results obtained from hydraulic measurements, and quote some examples where the method is of value. In a series of measurements of the discharge from a hydraulic machine, the following values were obtained of the relation of the head of water to the discharge as follows:—

Discharge (Q) in Cubic Feet per Minute.	Head (H) in Feet.
2·56	0·192
7·95	0·302
23·10	0·463
83·52	0·774

It is known that the relation between Q and H is of the form $Q = CH^n$. Determine the values of c and n from the measurements, and calculate the discharge for $H = 0·623$ foot.

9. Describe the construction and action of one form of hydraulic accumulator. Water at a pressure of 2,000 lbs. per square inch is supplied by a pump for working a hydraulic press, and a steam accumulator having a 6-inch ram and 30-inch stroke is connected to the pipe line to moderate the inequalities of pressure and the fluctuations of the supply. Calculate the size of the piston, if the effective steam pressure is 120 lbs. per square inch, and the loss due to friction of the packings of the accumulator is 6 per cent. Also determine the work which the accumulator can perform for one stroke of the piston.

10. Describe by aid of sketches the construction and mode of action of a low-lift centrifugal pump. Show that in a pump of this kind with no whirlpool chamber, the peripheral velocity V of the wheel is given by the formula $V_o = \sqrt{(2gH + u_o^2 \operatorname{cosec}^2 \phi)}$, where u_o is the radial outlet velocity, and ϕ is the angle of the blades at the periphery of the wheel. Show what effect the variation of the angle ϕ has upon the velocity of the wheel, and describe briefly the special advantages of pumps (*a*) with radial blades at outlet, (*b*) with re-curved blades.

11. Describe the construction and action of a Pitot tube, and show how you would use it to obtain the velocity of water at different points in the section of a water main. Sketch the probable form of the velocity curve for a main in which the mean velocity of water is, say, 5 feet per second (1) at a section in a straight length of pipe, (2) at a bend of moderate curvature, say, of 4 diameters.

12. Give a short account of the results of experiments on the dissipation of head due to bends, elbows and sudden changes of section in a pipe, and deduce an expression for the loss of head due to sudden enlargement in a pipe line. A 4-inch main conveying water at a velocity of 10 feet per second is suddenly enlarged to 6 inches in diameter. Calculate the loss of head due to the change of section.

October, 1909.

HYDRAULICS.

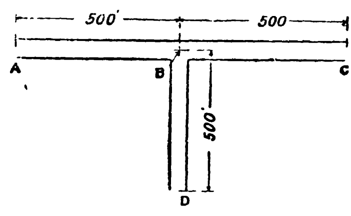
(*Not more than* EIGHT *questions to be attempted by any Candidate.*)

1. Twelve hundred gallons of water per minute are to be lifted a height of 150 feet through a pipe 1 foot diameter and 5,000 feet long. Assuming the head lost by friction in the pipe per foot length when the velocity is 1 foot per second is 0·0005 foot, and the mechanical efficiency of the pump 80 per cent., find the effective horse-power of the engine driving the pump.

2. The outlet circumference of a centrifugal pump has a diameter of 2 feet, and the inlet circumference is 1 foot 3 inches diameter. The radial velocity of flow at inlet is 5 feet and at outlet 8 feet per second. The wheel makes 400 revolutions per minute. The vanes are radial at exit. Determine the theoretical lift of the pump and the direction of the inlet edge of the vane.

3. Show that Bernoulli's theorem is consistent with the principle of the conservation of energy. One cubic foot of water per second flows along a pipe, which has a slope of 1 in 10, from A to B. At A the diameter is 3 inches and at B 6 inches, and the horizontal distance from A to B is 50 feet. The pressure head at A is 20 feet of water and at B 25 feet. Find, from this data, the head lost between A and B.

4. A turbine is driven by a head of 120 feet. The water used in a test of the turbine is measured by passing it through a right-angled V notch. The mean head over the apex of the notch during the trial is found to be 9 inches. The brake-horse-power of the turbine is found to be 13; find the efficiency of the turbine.



5. A horizontal pipe, A C in the figure, has a diameter of 6 inches and is 1,000 feet long. At point B a horizontal branch-pipe 3 inches diameter and 500 feet long is taken off A C. The pressures at A, C, and D are found to be 20 feet, 10 feet, and 7·5 feet of water respectively. Assuming the head lost by friction in a length l of either of the pipes to be $\frac{0{\cdot}0006\, l\, v^2}{d}$, and neglecting other losses, show that the velocity in A B is about 3·16 feet per second, and find the pressure at B.

6. Water issues from a square-edged vertical circular orifice 2 inches diameter, made in the side of a vessel in which a constant pressure of 200 lbs. per square inch is maintained, and the head of water in the vessel above the centre of the orifice is 8 feet. A flat plate is placed near to the orifice, the plate being inclined at an angle of 60° to the horizontal plane. Find the discharge from the orifice and the horizontal pressure on the plate.

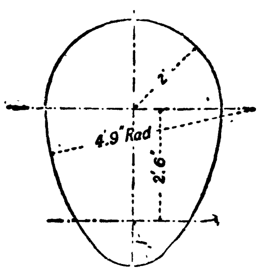
7. Describe briefly, with the aid of sketches, two methods of determining the discharge of a river 100 feet wide.

8. Make sketches of and explain the principle of action of one of the following:—(*a*) A compressed air pump showing the lift, and the depth of water in the well; (*b*) a hydraulic ram; (*c*) a hydraulic press in which the ram is returned automatically.

9. Make sketches of two arrangements for regulating the flow of water to pressure turbines and of one arrangement for regulating the flow to one form of impulse turbine. Discuss the effects of the arrangements in diminishing the efficiency of the turbines.

10. A hydraulic direct-acting lift has a ram 9 inches diameter. The pipe connecting the valve-box to the cylinder is $\frac{3}{4}$ inch diameter. The pressure in the valve-box is 600 lbs. per square inch. Neglecting all other losses than that due to sudden enlargement when the water enters the cylinder, and assuming the valve fully open at all loads, find the maximum load that can be lifted at a velocity of 1 foot per second. Find also the maximum velocity with which the lift could descend, assuming the exhaust full open.

11. A brick-lined sewer has a section shown in the figure, and a slope of 1 in 1,000. Determine the discharge of the sewer in cubic feet per second



when the depth of flow is 2 feet. Explain the formula you use. (*Note.*—Areas, &c., may be found from a scale drawing.)

12. Give an account of any experiments with which you are familiar on the discharge of water over weirs. State the deductions to which the experiments would lead you in making a weir for the accurate gauging of water, with respect to the following points:—(*a*) The form of the sill of the weir, (*b*) the form of the approaching channel, (*c*) any precautions to be taken in connection with the down-stream channel.

February, 1910.

HYDRAULICS.

Not more than EIGHT *questions to be attempted by any Candidate.*

1. The discharges over a weir under different heads were as follows:—

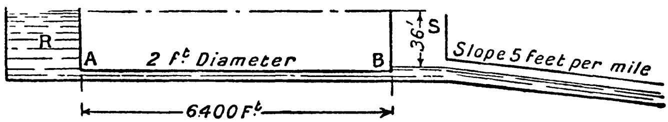
Head.	Discharge in Cubic Feet per Second.
0·5	11·5
1·0	33·3
1·5	61·5

On a certain day the following readings of the head over the weir were taken :—

Time.	Head.
2 o'clock	1·0
3 „	1·125
4 „	1·250

Plot a head-discharge curve for the weir, and also a time-discharge curve, and determine approximately the discharge over the weir between 2 o'clock and 4 o'clock.

2. A pipe A B, 2 feet in diameter, conveys water from a reservoir to a sump S. From S the water runs along a circular culvert having a slope of 5 feet per mile. When the difference of level of the water in the reservoir



and the sump is 36 feet, the water in the aqueduct is required to be flowing steadily at a level coinciding with the centre. Find the diameter of the culvert. The value of C in the formula $v = C\sqrt{m\,i}$ may be taken as 100 for the pipe and culvert.

3. Two vessels are connected by a horizontal pipe 3 inches diameter and 20 feet in length. The surface of the water in one vessel is kept at a height of 10 feet above the centre of the pipe, and the pressure above the surface is 14·7 lbs. per square inch absolute. The surface of the water in the second vessel is kept at 3 feet above the centre of the pipe, and the pressure above the surface is 2 lbs. per square inch absolute. The pipe is not bell-mouthed at either end. Find the discharge per second into the second vessel. The head lost by friction is $h = \frac{0 \cdot 04\, l\, v^2}{2\, g\, d}$.

4. The tanks of the last question are each 12 feet long by 12 feet wide. Assuming that the water in the first vessel is at any instant 10 feet above the centre of the pipe, and the surface of the water in the other is 3 feet above the centre, and that no water enters the first vessel while flow takes place into the second, and the pressure above the surfaces remains constant, find the time taken for the surfaces to come to the same level.

5. The plunger of a pump is 6 inches diameter, its stroke is 1 foot, and it makes 30 strokes per minute. The pump has an air vessel on the delivery. The mean pressure at the delivery valve of the pump is 700 lbs. per square inch. The pump delivers water to a crane having a ram 10 inches diameter through a pipe 1 inch diameter. The velocity ratio of the crane hook to the ram is 6. Neglecting all losses, except loss by sudden enlargement when the water enters the crane cylinder, find the maximum load the crane can lift. The mechanical efficiency of the crane is 85 per cent.

6. Sketch a Poncelet wheel, or some form of impulse turbine, and show clearly how you would determine the form of the vanes for maximum efficiency. Show that for a Poncelet wheel, neglecting friction, the most efficient velocity of the wheel rim is $v = \frac{1}{2}\cos\theta\sqrt{2\,g\,H}$. H = head. θ = the angle between the approaching water and the tangent to the wheel.

7. Make sketches showing the construction of either :—(*a*) Some form of high lift centrifugal pump, showing clearly the arrangement for con-

verting velocity head into pressure head, or (b) a reciprocating pump suitable for supplying water to hydraulic cranes at a pressure of 750 lbs. per square inch, showing clearly the arrangement of the valves, and the packings for the plunger and the glands.

8. Describe briefly, and show by sketches, how you would determine the loss of head in any length of a pipe of constant diameter along which water is flowing, and hence from a number of experiments how you would deduce a formula to express the loss of head at any velocity in, say, clean cast-iron pipes. Quote the formulæ you would be likely to get for (a) clean cast-iron pipes; (b) clean brass pipes.

9. Show that in a turbine the work done per lb. of water on the wheel is—

$$\frac{Vv}{g} - \frac{V_1 v_1}{g} = H - \frac{\mu^2}{2g}.$$

V = velocity of whirl at inlet;
V_1 = „ „ „ outlet;
v = „ „ wheel at inlet;
v_1 = „ „ „ outlet;
H = head under which turbine works;
μ = velocity of the water as it leaves the turbine.

10. A jet delivers 100 cubic feet of water per second under a head of 100 feet, and strikes a plane inclined at 45° to the direction of the jet. Calculate the normal pressure on the plane.

11. The inner and outer diameters of an inward flow turbine wheel are 1 foot 6 inches and 3 feet respectively, and the velocity of the wheel periphery is 45 feet per second. The radial velocity of flow is 8 feet per second. The water leaves the guide blades in a direction making 20° with the tangent to the wheel, and leaves the wheel radially. Determine the angles of the wheel vanes at both circumferences, and the head under which the wheel is working.

12. Make sketches of the following:—A hydraulic crane cylinder in which the quantity of water supplied for a given lift can be diminished when the load is small, or a hydraulic engine in which the quantity of water used is regulated according to the work done.

October, 1910.

HYDRAULICS.

Not more than* EIGHT *questions to be attempted by any Candidate.

1. A pair of sluice gates, with square orifices 3 feet by 3 feet, deliver a submerged flow into a lock chamber whose length from gate to gate is 300 feet with an average width of 40 feet. What time will it take to fill the chamber if the lift is 16 feet? Coefficient of discharge = 0·65.

2. Define the term "hydraulic gradient," and explain its practical use in designing a water main. A cast-iron pipe is 8 inches in diameter, and is

laid for 1 mile at a slope of 1 in 500, for another mile at a slope of 1 in 100, and for a third mile is level. The level of the water is 20 feet above the inlet end and 6 feet above the outlet. Find the discharge and draw the hydraulic gradient.

3. A tank discharges through a pipe $1\frac{1}{4}$ inches diameter. The pipe is turned up in a vertical direction at the end, its outlet being 30 feet below the level of the water in the tank. The water issues from the pipe, full bore, and rises to a height of 6 feet. What height would the water rise if the end of the pipe had been finished by a short smooth nozzle $\frac{1}{2}$ inch diameter of negligible resistance?

4. The velocity, hydraulic radius, and slope for a stream flowing in a rectangular channel 8 feet wide are connected by the formula $S = \mu \frac{V^2}{R}$. Draw curves showing the relative discharges when the depth of water is 1, 2, 3, 4, and 5 feet respectively. Show on the same diagram, for each depth, the maximum discharge that can be obtained from a rectangular channel of the same area of cross-section as the stream has for that particular depth.

5. Describe fully the apparatus you would use, and the actual operations in the field, when measuring the flow of a river about 150 feet wide, 15 feet greatest depth, with a mean surface velocity of about 4 feet per second.

6. Derive an expression for the velocity and pressure at any point of a free cylindrical vortex. Mention one or two examples where a free vortex occurs in practical hydraulic work.

7. A press required to give a maximum pressure of 200 tons is supplied with water at 800 lbs. per square inch from an accumulator. The usual rate of travel of the press ram is to be 2 inches per minute, and the maximum rate twice that value; but this will only be required for a distance not exceeding 1 inch. What type and size of accumulator would you instal? and what power pump?

8. Describe, with sketches, and explain the action of a hydraulic brake. Calculate the area of the orifices in a brake piston which has to exert a resistance of 40 tons when moving with a velocity of 10 feet per second, the effective area of the piston being 100 square inches. All frictional resistances may be neglected.

9. Water enters and leaves a rotating wheel without shock or frictional loss, and without change of pressure. Derive an expression connecting the velocities of rotation of the wheel at inlet and outlet with the relative velocities of the water to the wheel at those points.

10. Find an expression for the lift and efficiency of a centrifugal pump when no vortex chamber is provided. A pump has a lift of 20 feet and a radial velocity of flow at inlet and outlet of 9 feet per second. Calculate for vane angles at outlet of 90°, 45°, and 30°, the velocity of the wheel to maintain this head, and the efficiency in each case.

11. A Barker's mill is made with two nozzles, each rotating arm being 2 feet long. The nozzle diameters are $\frac{3}{4}$ inch, the revolutions of the wheel 120 per minute, and the head-producing discharge is 1 foot. Find the power developed and the efficiency, neglecting friction.

12. Make clear outline sketches, showing the valves, of a bucket pump, a single-acting plunger pump, and a piston and plunger pump.

February, 1911.

HYDRAULICS.

Not more than EIGHT *questions to be attempted by any Candidate.*

1. Write down the ordinary form of Bernoulli's theorem, and discuss the derivation of each term. Discuss the way in which it is modified to apply to practical hydraulic problems.

2. A tank discharges under a head of 10 feet through a short, sharp-edged, cylindrical tube 4 inches diameter. The coefficient of contraction is 0·62, and the velocity of discharge, full bore, at the end of the tube is 20·8 feet per second. Assuming that the whole loss of energy occurs between the contracted section and the outlet, calculate the pressure at the contracted section. Compare the discharge given above with that under the same head from a 4-inch re-entrant cylindrical mouthpiece and a 4-inch sharp-edged orifice.

3. Show, by means of sketches, how you would construct a sharp-edged rectangular weir to be used for gauging a stream discharging about 10 cubic feet per second. State the precautions to be taken in arranging its proportions and its position in the stream.

4. Find the head necessary to discharge 45 cubic feet per second through a culvert 2 feet 9 inches diameter and 80 feet long. The inlet and outlet are both cylindrical. The culvert is lined with smooth brick, for which $n = 1·75$, $m = 1·17$, and $\log \mu = 5·9$ in the formula $S = \mu \frac{V^n}{R^m}$. The water-level at the outlet being 2 feet 6 inches above the centre of the culvert, and the culvert horizontal, sketch the hydraulic gradient.

5. A 3-inch pipe suddenly enlarges to 6 inches. When the quantity of water passing is 0·75 cubic foot per second, the head lost at the change of section is found by experiment to be 2·16 feet. Calculate the head lost by the usual formula, and comment upon the difference in the value so found and the experimental value.

6. Describe, in a concise way, the nature of the difficulties to navigation presented by a river in its upper reaches where the low-water flow is small and the slope considerable, and in its lower reaches where the slope is small and the river winding.

7. A jet of water delivering 2,500 gallons per minute with a velocity of 30 feet per second is freely deviated through an angle of 30° by impinging on a series of vanes. The velocity of the jet after leaving the vanes is 10 feet per second. Find the magnitude and direction of the resultant force on a vane.

8. Water enters a radial-flow impulse turbine with a velocity V_1, having a radial component $N_1 = K V_1$. The velocity of discharge from the wheel is radial and equal to N_1. Find an expression for the velocity U_1 of the wheel at the inlet and the efficiency in terms of V_1 and K.

9. Describe with sketches some forms of turbine suitable for working on low falls when the quantity and head are subject to variation.

10. A single-acting plunger pump has a plunger diameter of 1 foot, a stroke of 2 feet, and makes 25 revolutions per minute. It is placed 20 feet above the level of the tank from which it is pumping, the length of the cast-iron suction pipe is 30 feet and its diameter 9 inches. Find

the pressure in the cylinder at (*a*) the commencement of the suction stroke, (*b*) the middle of the suction stroke. Assume the plunger has simple harmonic motion.

11. Give a list of the kinds of water meters in common use and make detailed sketches of one type, giving its special advantages and the circumstances under which you would use it.

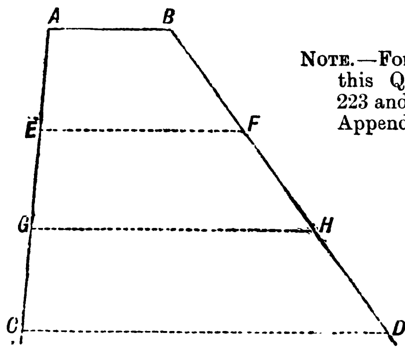
12. Describe briefly the important features of a scheme for the hydraulic transmission of power from a central station to private consumers. Deal with the central-station plant, the transmission pipe-lines, and the type of machinery most suitable for operation.

October, 1911.

HYDRAULICS.

Not more than EIGHT *questions to be attempted by any Candidate.*

1. The sketch A B C D shows the upper part of the section of a masonry dam: dimensions A B = 5 feet, C D = 15 feet, vertical height A C = 12 feet, front batter from A to C = 1 foot, or 1 in 12. Assume the masonry is of uniform density, 160 lbs. per cubic foot, and water to weigh $62\frac{1}{2}$ lbs. per cubic foot. Divide the depth into three equal 4-foot parts at the horizontal sections E F, G H. Sketch the graphic constructions whereby



NOTE.—For a solution of this Question, see p. 223 and the end of this Appendix.

SECTION OF THE UPPER PART OF A MASONRY DAM.

the centres of gravity of these three parts of the masonry can be found, and calculate their weights arithmetically. Sketch the graphic constructions whereby the centres of water-pressure on the face sections E G and G C when the water level is at E are to be found. Calculate arithmetically the magnitudes of these pressures, and the positions of the above centres of water-pressure upon them.

2. Compare the full-bore volumetric flows per second through long straight pipes and conduits of uniform section, (1) in respect to their

various lengths, (2) in respect to their various sizes, and (3) with respect to their various inlet and outlet heads, all being of the same sectional shape, but no one particular shape being assumed. Use the letter "m" for mean hydraulic depth, "A" for sectional area, "v" for linear velocity, and "V" for volumetric flow per second.

3. Compare the volumetric flow per second through a square orifice of 6-inch side in the horizontal floor of a tank under a head of 1 foot with that through a square orifice of the same size in the vertical side of a tank under the same head 1 foot measured to the centre of the square. Assume that the comparison is not affected by the *vena contracta* in either case. Make a similar comparison with a head of 2 feet.

4. What is the mean hydraulic depth of the water in a pipe of circular section of diameter D when the level of the water is at the centre of the pipe? How does the mean hydraulic depth vary from this when the level of the water rises a small height d above the centre, and when it falls a small height d below it?

5. A pipe of circular section of 30-inch diameter has a fall of 16 feet in a straight run of 2,000 feet. In the formula for loss of head in feet per foot run $i = f v^2/m$, its coefficient is 0·00009, the units being feet and seconds. What volumetric flow in cubic feet per second must there be through this pipe to maintain throughout its length the level of the water at the centre of the circular section?

6. Water at 90 lbs. per square inch absolute pressure flows full bore with a velocity of 4 feet per second through a pipe of square section and 15 inches square inside size, which has in it a right-angled bend of 2 feet radius measured to centre of pipe section. Calculate the total outward and inward pressures in the plane of the bend upon the outer and inner halves of the walls of the bend, assuming that the distribution of velocity is not disturbed by centrifugal forces but remains uniform across each section.

7. Sketch the vector diagram to be used in adjusting the angles of guide-blades and wheel-blades of water turbines. Explain what practical data are required for the construction of this diagram. Use this diagram to show how inefficiency arises from variation of the turbine speed. What item of inefficiency, apart from frictional resistances, is unavoidable?

8. Explain the action and the advantages of air-vessels placed (1) on the delivery side, and (2) on the section side of reciprocating pumps. What are the two main quantities to which the size of such air-vessels should be proportioned?

9. Describe in outline the construction of compound centrifugal water-pumps, and give sketches of one form of such pump without small detail.

10. State what you know about recent investigations of the disturbance of stream line motion by the method of insertion of colour bands in the flow of water along pipes and channels.

11. Describe the construction and action of a hydraulic-relay speed-governor for a large water turbine. Give sketches of one form of such governor. What do you consider to be a good coefficient of speed variation to be attained without great complication and cost in the governing mechanism?

12. Discuss the relative advantages of rope-lifting gear and direct water-pressure plunger mechanism for passenger-lifts worked by hydraulic power.

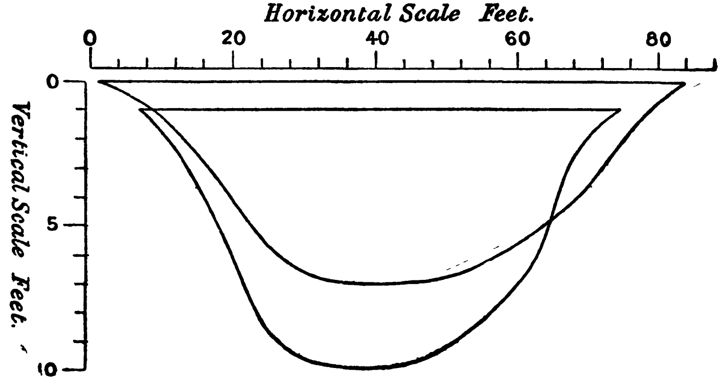
February, 1912.

HYDRAULICS.

Not more than EIGHT *questions to be attempted by any Candidate.*

1. Describe the current-meter which you think most suitable for measurement of linear velocity in open channels. Detail the construction, the mode of action, and a good method of calibration.

2. Describe the method you would adopt in gauging the flow in the river of which the appended scaled sketch gives two cross-sections,



200 yards apart, assuming that you are not restricted as to method by limitations of time or money. Between these sections the banks and bottoms are fairly straight-lined.

3. In the horizontal floor of a tank in which the water is 3 feet deep, and in which the water-surface is 15 square feet in area, a sharp-edged circular orifice 2 inches in diameter is opened. In what time will the water-level in the tank sink 1 inch, if there be no supply of water to the tank? In answering make no attempt at correction for the time spent in starting the flow—that is, upon initial acceleration of the mass of the first parts of the flow.

4. A straight pipe of circular section is 400 feet long and 4 inches internal diameter. Its down gradient is 1 in 200. It discharges full-bore into the atmosphere at the level of the lower end of the pipe, and draws by a bell mouth from a reservoir with 12 feet head of water over the mouth of the pipe. The gradient of the frictional loss of head in the pipe is 0·00012 times the square of the velocity divided by the mean hydraulic depth, the units being feet and seconds. Find the linear velocity through the pipe and the discharge in gallons per hour.

5. Assume that the loss of head in a round pipe L feet long and of diameter d feet is $4fv^2L/d$. The pipe has a down gradient i, and it discharges freely and full-bore into the atmosphere at the level of the

lower end of the pipe. It draws water from a reservoir in which the head over the inlet in the pipe is maintained constant and equal to h. Assume that there is no loss of head at the entrance into the pipe, or neglect this loss. Find the algebraic expression for the absolute pressure (in feet head of water from zero pressure) at the inlet to the pipe and draw a diagram showing the variation of water-pressure along the length of the pipe. Use the symbol h_0 for atmospheric pressure in feet of water. Could your formula for this inlet pressure ever show a zero or a negative value? If so, explain the physical reason for the resulting limit of applicability of the formula, and say what results in cases in which the formula shows a negative inlet pressure.

6. The waste weir of a dam is to discharge up to 700 cubic feet per second, with a maximum depth of water over the crest of 3 feet 6 inches. Take the side contraction at each end of weir as zero, and the coefficient in the ordinary Bazin formula, covering also the velocity of approach, as 0·041. What should be its width?

7. When a jet of water impinges obliquely upon a fixed, flat, smooth plate, how does the jet divide, and what is the pressure upon the plate in terms of the cross-sectional area, the linear velocity, and the obliquity of the jet?

8. Explain the main difference between the modes of action of the water upon the turbine blades of the two forms of turbine commonly termed "impulse" and "pressure."

9. Sketch and explain a gear for speed multiplication in hydraulic cranes for rapid lifting of light loads.

10. Discuss the relative advantages of hydraulic and hand riveting.

11. Give the mean hydraulic depths of circular, square, and regular octagonal sections, each of the area 5 square feet; and give also for each section the value of $\sqrt{m}$ which is used in the calculation of the water velocity in pipes.

12. Explain the term "critical velocity," and state what conditions determine its magnitude.

October, 1912.

HYDRAULICS.

Not more than EIGHT *questions to be attempted by any Candidate.*

1. Assuming that the discharge of a pipe, with hydraulic gradient, i, equal to 1 in 20 d, when d is the diameter of the pipe in inches, equals d^2 cubic feet per minute; give the coefficient in the Chezy formula for the velocity, $v = c\sqrt{d\,i}$, that the use of this rule is equivalent to.

2. Explain the difference between an impulse and a reaction turbine, and state the conditions most suitable for the employment of each type.

3. Describe a form of turbine pump in which the volume of water automatically increases as the head is reduced without materially increasing the power required. How is a centrifugal pump designed to deal with heads of over 100 feet, say?

4. A three-throw pump is used for pumping sewage, the diameter of the plungers being 2 feet and the stroke 3 feet, and the number of double strokes 20 per minute; find the delivery of the pump, allowing for reasonable slip. Also calculate the power of the engine necessary to work the pump if the total head, including friction, is 50 feet.

5. It is required to construct a main drain to collect the storm-water from a total length of 5 miles of roads, average width 30 feet. Assuming that the time it will take the rainfall to enter the subsidiary drains and to be carried to the main drain is 20 minutes, and that the maximum rainfall during that time is $\frac{1}{3}$ inch, also that 60 per cent. of the rainfall reaches the drain; find what quantity of water per minute the main drain will have to provide for.

6. Describe a form of hydraulic ram pump. If the efficiency of the ram is 70 per cent. and 10,000 gallons of water per hour be taken from a river with an effective fall of 20 feet, what quantity of water would the ram be able to lift to a height of 200 feet?

7. Sketch an arrangement of a rectangular weir across a stream for purposes of obtaining continuous gaugings of the stream, assuming that there is sufficient fall to permit of this being done.

8. Calculate the loss of head per mile in a pipe 24 inches diameter if the velocity is 4 feet per second.

9. Describe a form of differential gauge containing water and oil which may be used to measure the loss of head in a length of pipe.

10. Draw curves showing the variation of the velocity of flow of the water in horizontal and vertical planes at a cross-section of a river on a straight run; and explain how these would be affected at a bend.

11. Why should sudden enlargements or contractions of area in a pipe through which water is to flow be avoided if possible; what is the extent of the loss of head due to such variations in diameter, and how is the loss actually occasioned?

12. If water moving with velocity v, impinges at right angles on a plane surface area A moving with velocity V, what is the pressure on the plane and the efficiency of the jet, and what ratio must subsist between the velocity of the vane and that of the jet for the efficiency to be a maximum?

February, 1913.

HYDRAULICS.

Not more than EIGHT *questions to be attempted by any Candidate.*

1. Give a formula for the reduction of pressure at the throat of a Venturi meter in terms of the full and reduced areas of the pipe and the quantity of water flowing through it. Explain how a Venturi pipe can be utilised for mixing lime-water or other solution with water flowing through a pipe in proportion to its quantity.

2. Under what circumstances would a three-throw plunger pump be more suitable than a centrifugal pump for pumping sewage? Sketch a form of pump chamber that would be appropriate for the purpose.

3. If v is the velocity of the wind in miles per hour, and the pressure of the wind in lbs. on the vanes of the disc of a wind engine equals $0 \cdot 003\ v^2$, find an expression for the maximum limit of the energy expended on the wind engine by the wind. Assuming that the work done is 50 per cent. of this maximum, find the horse-power of the wind engine, if the diameter of the disc is 25 feet, and the sails occupy two-thirds of its area, the velocity of the wind being 15 miles per hour.

4. If a rectangular notch is 10 feet wide and a depth of 9 inches of water flows over it, what quantity of water would be passing in gallons per minute?

5. What precautions should be taken in measuring the height of water over the sill of a rectangular notch; and if a velocity of approach cannot be avoided, how can this be allowed for in calculating the discharge?

6. Describe a form of current meter and explain how you would by its means find the discharge of a river where crossed by a single-span bridge.

7. Draw a normal tide diagram for a tidal range of 13 feet. If the inlet to an outfall sewer is 4 feet below high water of such a tide, what is the minimum time that the sewage would have to be stored on that tide?

8. Under what circumstances is a triangular notch a suitable method of gauging a stream? Taking the formula for discharge in cubic feet per second, in the case that the angle subtended at the V is a right angle, as $\frac{5}{2} H^{\frac{5}{2}}$, where H is in feet, what would be the daily discharge of a stream if the depth of water in the notch (H) equals 6 inches?

9. Explain how the loss of head in water flowing through a pipe occurs due to a right-angle bend in a pipe, and how such a loss can be measured.

10. Water flows through a 15-inch diameter pipe from a reservoir with water-level 200 feet above the datum, for a distance of 3 miles, to a tank of which the water-level is 170 feet above the datum. Find the discharge of the pipe on the assumption that it nowhere rises above a line joining the water-levels in the reservoir and tank.

11. Explain why a bend pipe, in a pipe-line conveying water under pressure, requires packing to keep it in position. Work out a formula for the outward pressure at a bend in such a case.

12. Find what relation should exist between the velocity of the impinging jet and the velocity of the vanes in a Pelton wheel, for maximum efficiency.

October, 1913.

HYDRAULICS.

Not more than* EIGHT *questions to be attempted by any Candidate.

1. What should be the ratio of depth to breadth in a rectangular channel for the hydraulic mean depth to be a maximum? Such a channel discharges 90 cubic feet per second, with a mean velocity of 5 feet per second. C in Chezy's formula, $v = c\sqrt{m\,i}$, is 100. Determine its slope.

2. A pipe of varying diameter discharges 10 cubic feet per second. At one section the area is 1 square foot, and the pressure is 40 lbs. per square inch. At a second section, 40 feet below the first, the area is $0 \cdot 5$ square foot. If flow takes place from the first to the second section, and if the friction loss between the two is 2 feet, determine the pressure at the second section.

3. A jet issuing from an orifice 2 square inches in area, under a head of 100 feet ($C_v = 0{\cdot}98$), strikes tangentially a series of vanes moving at 60 feet per second in the same direction as the jet. These vanes are curved backwards through 161° 48′. Determine the force on the vanes in the direction of motion. (Cos 161° 48′ = − 0·9500.)

4. Deduce an expression for the rise in pressure which would accompany the "sudden" closure of a valve at the end of a rigid water-main. Under what circumstances may the closure of a valve be considered as being "sudden" in this sense?

5. A sharp-edged weir having no side contractions discharges between two parallel side walls. Describe, with sketches, the various modifications which may occur in the following sheet or "nappe," and state in a general way their effect on the discharge.

6. A wide-crested spillway gives a discharge $Q = Kb\sqrt{H^3}$ cubic feet per second, where $K = 2{\cdot}64$. Its length b is 200 feet. A freshet raises the level of water in the reservoir to 2 feet above the crest. If the surface area of the reservoir is 1 square mile, how long will it take to reduce the height by 1 foot?

7. Experiments show that, under certain conditions, the resistance to the flow of water through a given pipe-line is directly proportional to the mean velocity v; under other conditions to v^n, where n is nearly 2. Explain this, and give a brief outline of any experiments which have been carried out to investigate this point.

8. Describe, with sketches, any form of hydraulic accumulator. What part does it play in a hydraulic-power pumping station?

9. Describe two methods of governing either an impulse wheel or a pressure turbine, stating the advantages of, or the objections to, each.

10. A Pelton wheel is to develop 140 B.H.P. under an effective head (at the nozzle) of 100 feet. Its efficiency is 80 per cent. If the coefficient of velocity is 0·98, what is the required diameter of the nozzle?

11. In an inward-flow pressure turbine the guide vane angle α is 14° 5′, and the vane angle at entrance to the wheel is 90°. On leaving the guides the velocity of the water is 30 feet per second. The speed of the wheel is adjusted so as to give entry without shock, and the vane angle, γ, at exit from the wheel is designed to give radial discharge. If the wheel is so shaped as to make the radial component of the velocity of the water the same at entry and exit, and if its inner and outer radii are 3 feet and 2 feet respectively, what is the value of γ, and how many revolutions per minute does the wheel make? Cos 14° 5′ = 0·9750; sin 14° 5′ = 0·2433.

12. What is meant by a "free" vortex? In such a vortex, if the axis is vertical, and if the velocity is v_1, and the pressure p_1, at a distance r_1 from the axis, find the pressure at a point in the same horizontal plane at a distance $2r_1$ from the axis.

February, 1914.

HYDRAULICS.

Not more than EIGHT *questions to be attempted by any Candidate.*

1. A rectangular sluice-gate is to be balanced about an horizontal axis. The gate is vertical; its depth is 4 feet; and the depth of its centre below the free water surface is 18 feet. At what depth should be the axis?

2. Two reservoirs are coupled by means of two parallel pipe-lines of equal length, and of diameters d_1 and d_2 feet. If the coefficient of friction is the same for both pipes, what is the ratio of their discharges?

3. Describe the Venturi meter for the measurement of water, and deduce a formula for obtaining its discharge from a knowledge of the difference of heads at entrance and its at throat.

4. A pipe-line consists of 1 mile of 12-inch pipe, and of 2 miles of 6-inch pipe. The coefficient of friction is 0·008 for the larger and 0·01 for the smaller pipe. Assuming that the loss of head is proportional to the square of the velocity, and neglecting all but friction losses, determine the discharge under a head of 100 feet.

5. Two reservoirs are connected by a pipe of diameter d and length l. This forms a siphon whose highest point is h feet above the surface of the higher reservoir. Show how you would determine the greatest practicable length between the inlet and the vertex of the siphon for the latter to run full. What happens if the length of inlet leg exceeds this?

6. Make sketches of, and explain the principle of action of one of the following:—

(1) Hydraulic buffer stop.
(2) Air-lift pump.

7. A stream discharges 150 cubic feet per second, and originally had a uniform depth of 2 feet and a width of 30 feet. A weir, 8 feet high, is built across the stream. The discharge over this weir is given by $3{\cdot}0\, b \sqrt{H^3}$ cubic feet per second. What is the depth of water at the up-stream side of the weir?

8. Describe, with the aid of diagrams, the phenomenon of "cavitation" or "separation" in a reciprocating pump not fitted with a suction air-vessel, when driven at too high a speed.

9. An accumulator has an 18-inch ram and 24 feet lift, and is loaded with 129 tons total weight. If packing friction accounts for 2·5 per cent. of the total pressure on the ram, determine the horse-power delivered to the mains if the ram falls steadily in 3 minutes, and if at the same time the pumps are delivering 624 gallons per minute to the accumulator.

10. The velocity of flow in a pipe-line is to be reduced from v_1 to v_2 feet per second by an increase in the section of the pipe. Deduce an expression for the consequent loss of energy if the change of section is sudden. If a taper pipe is used, with sides diverging at an angle θ, draw a curve showing approximately to scale how you would expect the loss to vary as θ increased from zero up to 180°.

11. Assuming the v^2 law of fluid friction, determine the horse-power necessary to drive a thin disc at 12,000 revolutions per minute if the disc is 9 inches in diameter and is totally submerged. Take the frictional resistance of a plane surface similar to that of the disc as 0·4 lb. per square foot at a velocity of 10 feet per second.

12. The peripheral speed of a centrifugal pump is 30 feet per second. The vanes are curved backward, so that the discharge angle γ is 35°, while the water leaves the impeller with a radial velocity of 5 feet per second. If the discharge is 120 cubic feet per minute, and the radius of the impeller is 2 feet, what is the hydraulic turning moment on the shaft? Take cot 35° = 1·428.

October, 1914.

HYDRAULICS.

Not more than EIGHT *questions to be attempted by any Candidate.*

1. Describe any means you are acquainted with for extracting water from a stratum at a considerable depth below the surface which contains fine sand mixed with coarser material, by means of a well or boring, so as to avoid trouble with sand when pumping.

2. For a drainage area of 1,000 acres, where a run-off of 1 inch per hour is to be expected, what length of overflow weir would you provide, if the depth of water over it is not to exceed 2 feet ?

3. In an air-lift water-raising plant, if the lift above the working water-level is 200 feet, about what depth should the air-pipe extend below the surface of the water for maximum efficiency ? If the air-pipe is submerged 200 feet below the working water-level, what pressure of air would be necessary at the bottom of the submerged pipe ?

4. Storm water has filled a storage culvert of square cross-section, built horizontal, during the rise of a tide with a range of 15 feet. The culvert is emptied by a pipe, 12 inches diameter and 2,000 feet long, laid to low-water level on the foreshore. The top of the culvert is at high-water level, its length is 500 feet, and its depth is 6 feet. Indicate how you would proceed to determine approximately the time required to empty the culvert. Take the tide-curve as being harmonic.

5. A pipe 12 inches diameter conducts water from an impounding reservoir to a service reservoir, the pipe being everywhere below the hydraulic gradient. If its length is 1,000 feet and the fall is 20 feet, find the rate of discharge. Why would the pipe have to be larger if the town were supplied direct from the impounding reservoir without the help of a service reservoir ?

6. Explain how to gauge a stream by means of current meters, also how to work out the discharge from the observations taken.

7. Sketch a bucket and plunger pump and explain why the variation in the rate of delivery is less with this type of pump than with an ordinary bucket pump.

8. Explain the cause of the losses of head which occur at sudden alterations in cross-section of a pipe flowing full. To what cause is the additional loss of head at a bend due ?

9. Sketch a pressure reducing valve. What is the difference between a break pressure tank and such a valve ?

10. Indicate how to design the shape of the vanes in a Poncelet undershot water-wheel.

11. Develop an expression for the efficiency of a turbine in terms of the tangential velocity of the water at the inlet, the velocity of the wheel at that point and the available head of water.

12. Explain with a sketch any way in which the efficiency in a centrifugal pump may be increased by the gradual reduction of the velocity of the water after leaving the vanes of the disc.

February, 1915.

HYDRAULICS.

Not more than EIGHT *questions to be attempted by any Candidate.*

1. Define what is meant by hydraulic gradient. A pipe flowing full conveys a quantity of water Q; at half its length a quantity $\frac{Q}{2}$ is taken off by a branch pipe, and the pipe is continued one-half the diameter of that of the first half of the length. Assuming the hydraulic gradient of the first half, draw that of the second.

2. The sewage of a town is discharged to sea on the first part of the ebb-tide. During the flood-tide it is collected in a storage culvert. Sketch a chamber to form a suitable connection between the storage culvert and the outlet-pipe to low-water mark, so that the culvert may empty in the shortest possible time.

3. A Venturi meter is fixed on a pipe 12 inches in diameter. The area at the neck of the meter is one-fifth that of the full-sized pipe. If the velocity in the full section is 4 feet per second, find the difference between the pressure in the full-sized pipe at the inlet and at the neck of the meter.

4. Explain how a small stream can be gauged by means of a right-angled V-notch. If the depth in the notch is 6 inches, find the discharge, using the formula $Q = \frac{5}{2} d^{\frac{5}{2}}$ cubic feet per second.

5. Illustrate by a sketch how a three-throw pump can be arranged in a well so that it would be always possible to gain access to the valves, even if it were drowned.

6. If a district comprises three or more basins, to the lowest points of which the sewage is drained, explain how the sewage could be lifted by compressed air by the use of a single air-compressing station.

7. The sewage of a town is delivered into a gauge-basin, with a rectangular notch leading to settling tanks and a rectangular notch leading to the storm-tanks. There are 10,000 inhabitants in the town, and the consumption of water is 25 gallons per head per day. If the sewage must not be allowed to pass to the storm-tanks until three times the dry-weather flow is passing to the settling-tanks, what height would you place the sill of the overflow weir to the storm-tanks with reference to that of the other weir, if the latter is 18 inches wide?

8. Assuming that the efficiency of a ram for lifting water is 75 per cent., what flow of water, with a fall of 10 feet, is necessary to deliver 9,000 gallons per day to a height of 200 feet?

9. Explain the principle of regulating the speed, according to the power required, in an axial flow and in a radial inward flow-reaction turbine.

10. Develop the formula for the increase in pressure due to a forced vortex. How is this made use of in the design of a centrifugal pump?

11. What is meant by water-hammer in pipes, and what considerations regulate the pressure developed if a valve at the bottom of a long length of pipe through which water is flowing is suddenly closed? Indicate how to proceed to calculate the increase in pressure so caused.

12. A pipe 12 inches in diameter has a hydraulic gradient of 1 in 500. What quantity of water will it discharge? If two pipes of equal diameter are to be used instead of the one pipe, and the same quantity of water is to be delivered, what diameter must they be?

October, 1915.

HYDRAULICS.

Not more than EIGHT *questions to be attempted by any Candidate.*

1. What is the statical pressure on one side of a circular area 3 feet in diameter, immersed vertically with its centre at a depth of 6 feet? Where is its Centre of Pressure?

2. Explain, with sketches, how the motion of a steady stream is affected during flow around a bend. How does this account for the position of the detritus at a bend in a natural stream?

3. The model of a broad crested weir is found to give a discharge, $Q = 3 \cdot 0\, bh^{\frac{3}{2}}$ cubic feet per second. What should be the length of a similar weir to give a discharge of 1,000 cubic feet per second under a head of 2 feet? How long would it take to lower the level of a basin 10,000 square feet in area, from 2 feet to 1 foot, over such a weir?

4. A Venturi meter has a main diameter of 4 feet and a throat diameter of 1 foot. When discharging 20 cubic feet per second the throat pressure is 10 feet of water. What is the pressure at entrance if the meter is fixed vertically with the entrance uppermost, the length from entrance to throat being 10 feet?

5. The water in a forced vortex makes 60 revolutions per minute. What is the difference of surface level at two points of radii 2 feet and 1 foot?

6. What are the advantages and disadvantages of duplicating the pipe line of a water-supply system. In such a pipe line how would you guard against the ill-effects of (*a*) an accumulation of air; (*b*) water-hammer?

7. Sketch and describe either (*a*) some form of hydraulically-operated penstock or gate valve, or (*b*) the essential features of some form of hydraulic crane.

8. Sketch and describe some form of automatic device by means of which a fairly constant flow may be delivered to an irrigation ditch, from a canal whose surface-level may vary within wide limits.

9. A canal of trapezoidal section has side slopes of 3 horizontal to 1 vertical; a bottom width of 10 feet; and a central depth of 5 feet. The value of C in Chezy's formula is 100. What is the discharge if the gradient is 1 in 1,000?

10. A 12-inch pipe line terminates in a conical nozzle giving a parallel jet 2 inches in diameter. The pressure in the main immediately behind the nozzle is 50 lbs. per square inch. What is the discharge if the coefficient of velocity is unity?

11. Experiment shows that there is always a loss of energy at a sudden enlargement of section in a hydraulic pipe line. To what is this due? Deduce an expression for its magnitude in terms of the velocity of flow in the smaller branch of the pipe.

12. Explain the meaning of the terms:—"Hydraulic gradient"; Standing Wave; Depressed Nappe; Cippoletti Weir.

February, 1916.

HYDRAULICS.

Not more than EIGHT *questions to be attempted by any Candidate.*

1. Explain the meaning of the terms "centre of pressure"; "water hammer"; "virtual slope"; "suppressed weir."

2. The impeller of a centrifugal pump has an external diameter of 12 inches and an internal diameter of 6 inches. If full of water, with the discharge pipe closed, what would be the difference of pressures at the outer and inner periphery, corresponding to a speed of 150 revolutions per minute?

3. Describe the construction and operation of a Pitot tube. How could you use it to measure velocities of flow in a pipe line?

4. A jet delivers 10 cubic feet of water per second under a head of 100 feet, and strikes a plane inclined at 45° to the direction of the jet. Assuming the coefficient of velocity to be unity, calculate the normal force exerted on the plane.

5. An 18-inch pipe line terminates in a junction box from which two pipe lines of equal length, and respectively 12 inches and 6 inches diameter, deliver the water into a common reservoir. If the surfaces of these pipes are equally rough, what will be their respective discharges when the 18-inch pipe line is delivering 10 cubic feet per second?

6. Describe the various devices adopted with a view of converting the kinetic energy of discharge from the impeller of a centrifugal pump into pressure energy. State generally under what circumstances you would think it advisable to use each of these devices.

7. Describe in detail how you would arrange a weir for measuring the discharge of a small stream. What precautions would you adopt to ensure accurate results?

8. The velocity of flow around a right-angled bend in a 12-inch pipe line is 15 feet per second. The pressure in the pipe line at this point is 20 lbs. per square inch. What is the resultant force tending to move the bend?

9. The available head in a Pelton wheel installation is 200 feet. The pipe line is 1,500 feet long and 36 inches diameter, and the nozzle gives a parallel jet 4 inches diameter. If the frictional coefficient f in the formula $h = \frac{f\,l\,v^2}{2g\,m}$ is 0·01, what is the horse-power of the jet?

10. What is meant by Bernouilli's Theorem? How do you use this to deduce an expression for the discharge from a Venturi meter in terms of the pipe areas and the velocity of flow?

11. A semicircular culvert is 4 feet in diameter and 1,000 feet long. If the constant C in Chezy's formula is 80, what will be the difference in level of the two ends when the culvert is discharging 60 cubic feet per second?

12. Sketch and describe briefly any two methods of regulating the supply of water to a pressure turbine.

City and Guilds of London Institute.

DEPARTMENT OF TECHNOLOGY.

TECHNOLOGICAL EXAMINATIONS.

46.—MECHANICAL ENGINEERING.*

HONOURS GRADE (Written Examination).

INSTRUCTIONS.

The Candidate for Honours must have previously passed in the Ordinary Grade, and is required to pass a Written and Practical Examination. He is requested to state, on the Yellow Form, whether he has elected to be examined in A, Machine Designing, or in B, Workshop Practice (*a*) *Fitting*, (*b*) *Turning*, (*c*) *Pattern making*. Candidates in Machine Designing must forward their work to London not later than May 6th,† and in Workshop Practice not later than May 13th.†

The number of the question must be placed before the answer in the worked paper.

The Candidate is at liberty to use divided scales, compasses, set squares, calculators, slide rules, and mathematical tables.

Five marks extra will be awarded for every answer worked out with the slide rule, provided the method of working is explained.

The maximum number of marks obtainable is affixed to each question.

Three hours allowed for this paper.

The Candidate is not expected to answer more than *eight* of the following questions, which must be selected from *two* sections *only*.

To obtain the certificate in Honours, the candidate must pass a Written and a Practical Examination, to be taken in the same year.

---

* The Questions for the Ordinary Grade are printed at the end of my *Elementary Manual on Applied Mechanics*.

† These dates are only approximate and subject to a slight alteration each year.

(1.) Written Examination.—In the Written Examination on *The Mechanics of Engineering*, candidates must select questions from not more than *two* of the following three divisions:—

(*A.*) The elasticity and strength of materials, including the more practical and elementary problems in compound stress. Tension, compression, and torsion. Combined bending and torsion. Combined thrust and bending. Riveted joints and the design of riveted work. Collapse. Behaviour of materials when tested. Ordinary limits of working stress (see the Author's Vol. II. on *Strength and Elasticity of Materials*).

(*B.*) The theory of the steam engine, including the thermodynamics of the action of steam. The solution of problems relating to the simpler valve gears. Governors and flywheels. The theory of gas engines, oil engines, and hot-air engines (see the Author's *Text-Book of Steam and the Steam Engine, including Turbines and Boilers*).

(*C.*) Hydraulics and hydraulic motor. Theory of flow from orifices. Flow in pipes. Water wheels, turbines and pumps. Construction and action of valves. Governors for hydraulic machinery. Hydraulic transmission of power. Hydraulic pressure engines. Lifts (see the Author's Vol. IV. on *Hydraulics*).

The Written Examination will be held in April of each year.

1909—TECHNOLOGICAL EXAMINATIONS.

C. & G.—46. MECHANICAL ENGINEERING.

HONOURS GRADE (Written Examination).

The Candidate is not expected to answer more than *eight* of the following questions, which must be selected from *two* sections *only*.

SECTION C (HYDRAULICS).

1. Prove that the discharge Q from a triangular notch is expressed by the formula—

$$Q = \frac{4}{15} c \sqrt{2g} B H^{\frac{3}{2}},$$

where H is the head of water, B is its surface breadth, and c is a constant derived from experiment. A right-angled notch is used to gauge the flow from a channel, and the average head is found to be 9·32 inches. Calculate the discharge per second if the coefficient of the weir is 0·617. (C. & G., 1909, H., Sec. C.)

2. Describe the principle of action of the Venturi water meter, and obtain a formula for the discharge through it. A meter of this form in a 12-inch water main has a throat 5 inches in diameter and the difference of head between the upstream section and the throat is 21 feet. Calculate the discharge from the meter per second on the assumption that its coefficient is unity. (C. & G., 1909, H., Sec. C.)

3. Describe the principal features in a modern hydraulic power station for distributing water at a high pressure for use in operating lifts, cranes, hydraulic presses, and the like. (C. & G., 1909, H., Sec. C.)

4. The ram of a hydraulic press, using water at 2,000 lbs. per square inch, has a diameter of 1 foot and a stroke of 8 inches. It is operated by an intensifier, which latter is supplied with water from the town main at a pressure of 30 lbs. per square inch. Sketch to scale a suitable form of such an intensifier to give a full stroke of the ram for a single stroke of the intensifier and mark on it the principal dimensions, explaining how you have determined them. (C. & G., 1909, H., Sec. C.)

5. A centrifugal pump is required to lift 800 gallons of water per minute through a height of 20 feet. Briefly describe, by aid of sketches, the construction of a pump suitable for this head and delivery, and, assuming all the formulæ necessary, calculate the principal dimensions of the pump and mark them on your sketches. (C. & G., 1909, H., Sec. C.)

6. Assuming the loss of head h in a cast-iron main of length l and diameter d to be expressed by—

$$h = \frac{\cdot 0004\, v^{1\cdot 87}}{d^{1\cdot 4}}$$

in feet and second units, v being the velocity of the water, determine the loss of head due to frictional resistances in a cast-iron water main 2 feet in diameter and 2 miles in length if the mean velocity is 4 feet per second. (C. & G., 1909, H., Sec. C.)

7. Describe, by aid of sketches, the construction and mode of action of one form of multi-stage turbine pump, and explain what features of turbine design have been successfully introduced to obtain a very high lift. (C. & G., 1909, H., Sec. C.)

1910—TECHNOLOGICAL EXAMINATIONS.

C. & G.—46. MECHANICAL ENGINEERING.

Honours Grade (Written Examination).

The Candidate is not expected to answer more than *eight* of the following questions, which must be selected from *two* sections *only* (A, B, C).

Section C (Hydraulics).

15. Describe, by aid of sketches, the construction and mode of operation of a governor gear suitable for regulating the speed of a large hydraulic turbine. (C. and G., 1910, H., Sec. C.)

16. Explain what are the chief advantages of rectangular notches for the measurement of the flow of water. Obtain an expression for the discharge Q per second over a sharp edged weir of width L in the form—

$$Q = \alpha (L - \beta H) H^{\frac{3}{2}},$$

and explain clearly the reasoning used to establish the formula. Give the numerical values of α and β if the discharge is expressed in cubic feet per second and the linear measurements are in feet. (C. and G., 1910, H., Sec. C.)

17. A hydraulic press, having a ram with a 6-inch stroke, is required to exert a total pressure of 200,000 lbs., using an available water pressure of 2,500 lbs. per square inch. There are two supporting columns for the head which is 3 feet above the lowest position of the ram. Determine the principal dimensions of the press, excluding the head, and show clearly what data you assume and the formulæ you use to obtain your results. Sketch the ram and cylinder to scale in longitudinal section.
(C. and G., 1910, H., Sec. C.)

18. Describe the construction and action of a Pelton wheel of, say, 200 horse-power suitable for a water supply at a pressure of 300 lbs. per square inch. Find an expression for the theoretical efficiency of such a wheel at any speed, and draw a curve showing this efficiency for all possible speeds of the buckets. (C. and G., 1910, H., Sec. C.)

19. Describe, by aid of sketches, the principal hydraulic features of construction of a canal lift for transferring barges from one level to another.
(C. and G., 1910, H., Sec. C.)

20. Describe the construction and action of a low lift centrifugal pump, and prove that its theoretical efficiency, with radial blades at outlet, is not greater than 50 per cent. Explain what advantages are obtained by curving the blades at outlet, and find an expression for the efficiency of a pump with blades of this form. (C. and G., 1910, H., Sec. C.)

21. Describe one form of modern pump used either for (i.) lifting water from deep boreholes or for (ii.) the drainage of mines.
(C. and G., 1910, H., Sec. C.)

1911—TECHNOLOGICAL EXAMINATIONS.

C. & G.—46. MECHANICAL ENGINEERING.

Honours Grade (Written Examination).

The Candidate is not expected to answer more than *eight* of the following questions, which must be selected from *two* sections *only*.

Section C (Hydraulics).

15. Explain the meanings of the terms "centre of buoyancy" and "metacentre" for a floating body. Show that the distance between them is I/V, where V is the volume displaced by the floating body and I is the second moment of area of the water-line section. A floating stage, in the form of a closed box 40 feet long, 8 feet broad, and 5 feet deep, weighs 18 tons. What is the depth of immersion? Find the righting moment when the stage tilts through an angle of 10°. Sine, 10° = ·1736.
(C. & G., 1911, H., Sec. C.)

16. Describe, by aid of sketches, the principal features of construction and operation of a movable 10-ton hydraulic crane suitable for unloading goods directly from a ship into railway trucks, which are brought alongside by a line of rails directly underneath the crane. (C. & G., 1911, H., Sec. C.)

17. Describe. in some detail, how you would proceed to determine the discharge of a stream of about 15 feet width, 4 feet deep, and a maximum velocity of $1\frac{1}{2}$ feet per second. Give particulars of the apparatus you would use, and the way in which you would reduce your observations. (C. & G., 1911, H., Sec. C.)

18. Obtain an expression for the work done in rotating a disc immersed in water, assuming that the resistance R, for a metal surface of area S and moving with a velocity V through still water, is expressed by the relation $R = fSV^2$, where f is a coefficient. Calculate the horse-power required to rotate a brass disc, 12 inches in diameter, at 1,200 revolutions per minute, if the value of R for an immersed surface 1 square foot in area, and moving at the rate of 10 feet per second, is 0·25 lbs. (C. & G., 1911, H., Sec. C.)

19. Describe the principal features of construction of one form of hydraulic forging press capable of exerting a squeeze of 2,000 tons or more. (C. & G., 1911, H., Sec. C.)

20. Obtain an expression for the effective work done by a jet of water A square inches in sectional area, and issuing from a nozzle at a velocity of V feet per second, if the jet impinges on the vanes of a Pelton wheel, which, for the purposes of calculation, may be taken as a series of hemispherical cups moving at a velocity of v feet per second in the same direction as the jet. Calculate the useful horse-power exerted if the jet is 2 square inches in area, V is 160 feet per second, v is 70 feet per second, and the efficiency of the vanes is 85 per cent. (C. & G., 1911, H., Sec. C.)

21. Describe the construction and mode of action of a low-lift centrifugal pump. In a pump with no whirlpool chamber, show that, if the whole variation of pressure in the pump disc balances the lift H, the efficiency η of the pump is

$$\eta = (V_0^2 - u_0^2 \operatorname{cosec}^2 \phi)/2(V_0^2 - V_0 u_0 \cot \phi),$$

where V_0 is the peripheral velocity of the pump disc, u_0 is the radial outlet velocity of the water, and ϕ is the inclination of the blades at outlet. (C. & G., 1911, H., Sec. C.)

1912—TECHNOLOGICAL EXAMINATIONS.

C. & G.—46. MECHANICAL ENGINEERING.

Final or Honours Grade (Written Examination).

The Candidate is not expected to answer more than *eight* of the following questions, which must be selected from *two* sections *only.*

Section C (Hydraulics).

15. Obtain an expression for the time required to fill a canal lock. In a ship canal lock, 900 feet long and 65 feet wide, the difference of level is 16 feet. Two culverts are provided for filling the lock, each 10 feet high and 6 feet wide. Calculate the time required to fill the lock. (C. & G., 1912, H., Sec. C.)

16. Describe, by aid of sketches, approximately to scale, the construction of a hydraulic riveting machine suitable for riveting together the shell

plates of a locomotive boiler. Explain, and give reasons for, the special advantages of hydraulic riveting machines. (C. & G., 1912, H., Sec. C.)

17. Describe a form of apparatus suitable for registering the varying head of water flowing over a notch. The circulating water from a condenser is passed over a rectangular notch 1 foot wide, and the mean head is found to be 6·7 inches. Calculate the discharge per second if there is no coefficient of contraction. (C. & G., 1912, H., Sec. C.)

18. Describe how you would proceed to determine by experiment, the frictional loss of head due to the velocity of flow of water in a pipe. A 4-inch main, running along a factory floor, is supplied by a roof tank 75 feet above. Determine the head of water at a point in this line of piping 200 feet away from the supply tank if the loss of head h, due to friction and bends, is given by

$$h = 0{\cdot}0006\,\frac{l}{d}\,v^{1\cdot75},$$

the units being in feet and seconds and the velocity (v) of flow 5 feet per second. (C. & G., 1912, H., Sec. C.)

19. Describe the principal features of construction and action of a turbine pump suitable for a lift of 300 feet and of 500 gallons capacity per minute. (C. & G., 1912, H., Sec. C.)

20. A differential accumulator to work at 1,000 lbs. pressure is required to give 400 cubic inches of water per stroke, which latter may be from 48 to 60 inches. Calculate the main dimensions of this machine and mark them on a vertical section drawn to a convenient scale. (C. & G., 1912, H., Sec. C.)

21. Describe the principal features, including the safety devices, of a hydraulic passenger lift suitable for use in an hotel, the maximum number of people to be conveyed being twelve. (C. & G., 1912, H., Sec. C.)

1913—TECHNOLOGICAL EXAMINATIONS.

C. & G.—46. MECHANICAL ENGINEERING.

HONOURS GRADE (Written Examination).

The Candidate is not expected to answer more than *eight* of the following questions, which must be selected from *two* sections *only*.

SECTION C. (HYDRAULICS).

15. A vertical rectangular sluice is 6 feet wide and 4 feet deep. Its top edge is immersed at a depth of 4 feet. If the sluice be hinged at this edge, find the force required at the bottom edge to keep it closed. (C. & G., 1913, H., Sec. C.)

16. Describe, with sketches, the Venturi meter as arranged for readings on a mercury U gauge. In such a meter, with horizontal axis, the ratio of pipe area to throat area is 9 : 1. If the pressure at the throat is 20 feet of water, what is the pressure at entrance to the meter when the velocity of flow in the main is 5 feet per second? Neglect friction. (C. & G., 1913, H., Sec. C.)

17. Explain what is meant by "critical velocity" in fluid motion. In the case of water, how does fluid friction depend on (*a*) velocity, (*b*) density, (*c*) temperature, and (*d*) roughness of surface, at velocities respectively above and below the critical? (C. & G., 1913, H., Sec. C.)

18. A common formula for the friction loss in long pipes is

$$h = \frac{f l v^2}{2 g m}.$$

Explain the meaning of these terms. Assuming in this formula that $f = 0{\cdot}01$ where the unit of length is the foot, find the greatest horse-power that can be transmitted through a 3-inch pipe 5,000 feet long if the inlet pressure is maintained constant at 750 lbs. per square inch.

(C. & G., 1913, H., Sec. C.)

19. In the centrifugal pump, the water leaving the impeller usually has a very high velocity. Show how this velocity is affected by changes in the vane angles, and describe two methods by which a proportion of the kinetic energy of discharge may be converted into pressure energy before the water leaves the pump. (C. & G., 1913, H., Sec. C.)

20. In an inward-flow pressure turbine the lead-on angle of the guide vanes is 20°; the velocity of efflux from the guides 45 f.s., and the peripheral velocity of the wheel at entrance 32 f.s. Determine, either graphically or by calculation, the necessary runner-vane angle for entry without shock. If the ratio of inner to outer radius of the runner is 2 : 3, and if the width of the wheel is constant and the vane angle at discharge is 30°, determine the velocity of the water at exit. (C. & G., 1913, H., Sec. C.)

21. What is meant by "cavitation" in a reciprocating pump? A plunger moves with simple harmonic motion and draws water through a suction pipe of its own diameter and 100 feet long. The level of water in the suction tank is 14 feet below the pump. If the barometric height is 34 feet of water, how many revolutions per minute may the pump make without cavitation occurring at the beginning of the suction stroke?

(C. & G., 1913, H., Sec. C.)

1914—TECHNOLOGICAL EXAMINATIONS.

C. & G.—46. MECHANICAL ENGINEERING.

HONOURS GRADE (Written Examination).

The Candidate is not expected to answer more than *eight* of the following questions, which must be selected from *two* sections *only*.

SECTION C. (HYDRAULICS).

15. A vertical trapezoidal area is submerged with its upper edge horizontal and at a depth of 2 feet. The area is 4 feet wide at the top, and 2 feet at the bottom, and is 6 feet deep. At what point is its centre of pressure? (C. & G., 1914, H., Sec. C.)

16. Show how the Venturi meter is used for measuring the flow of water through a pressure main. Deduce an expression for the discharge through

such a meter, in terms of its dimensions. Would you require to modify this expression for use in practice, and if so, how?

(C. & G., 1914, H., Sec. C.)

17. Obtain an expression for the flow over a right-angled V notch under a given head *h* feet. State clearly what assumptions you make. Adopting what you consider suitable coefficients, determine the discharge under a head of 1 foot. (C. & G., 1914, H., Sec. C.)

18. Explain briefly the special difficulties which arise in regulating the speed of a water turbine, and describe, with the aid of sketches, the essential features of one form of regulating gear suitable for an inward-flow pressure turbine. (C. & G., 1914, H., Sec. C.)

19. A parallel pipe line of clean cast-iron piping is 12 inches in diameter and 2,000 feet long. It connects two reservoirs, the difference of whose surface levels is 25 feet. Use such coefficients as you think applicable to the case and determine the discharge in cubic feet per second.

(C. & G., 1914, H., Sec. C.)

20. Enumerate the various sources of loss in a centrifugal pump, and explain how these severally may be minimised. (C .& G., 1914, H., Sec. C.)

21. The following data were obtained from a test of a Pelton wheel :—

Area of jet, ...	11·0 square inches.
Discharge, ...	5·84 c.f.s.
Head at nozzle, ...	97·0 feet.
Brake horse-power, ...	50·0
H.P. absorbed in mechanical friction and windage, ...	2·7

Draw up a complete balance sheet showing the distribution of the energy of the supply water. Given the moment of inertia of the rotating parts, how would you suggest obtaining the energy absorbed in mechanical friction? (C. & G., 1914, H., Sec. C.)

THE CENTIMETRE, GRAMME, SECOND, OR C.G.S. SYSTEM OF UNITS OF MEASUREMENT AND THEIR DEFINITIONS.*

I. Fundamental Units.—The C.G.S. and the practical electrical units are derived from the following mechanical units :—

The *Centimetre* as a unit of *length*; the *Gramme* as a unit of *mass*; and the *Second* as a unit of *time.*

The *Centimetre* (cm.) is equal to 0·3937 inch in length, and nominally represents one thousand-millionth part, or $\frac{1}{1,000,000,000}$, of a quadrant of the earth.

The *Gramme* (gm.) is equal to 15·432 grains, and represents the mass of a cubic centimetre of water at 4° C. Also, 1 lb. of 16 ozs. is equal to 453·6 grammes. *Mass* (M) is the quantity of matter in a body.

The *Second* (s) is the time of one swing of a pendulum making 86,164·09 swings in a sidereal day, or the $\frac{1}{86,400}$ part of a mean solar day.

II. Derived Mechanical Units.—

Area (A or cm.^{2}).—The unit of area is the *square centimetre.*

Volume (V or cm.^{3}).—The unit of volume is the *cubic centimetre.*

Velocity (v or cm./s) is rate of change of position. It involves the idea of direction as well as that of magnitude. *Velocity* is *uniform* when equal distances are traversed in equal intervals of time. The unit of velocity is the velocity of a body which moves through unit distance in unit time, or the *velocity of one centimetre per second.*

Momentum (M v, or gm. × cm./s) is the quantity of motion in a body, and is measured by mass × velocity.

Acceleration (a or cm./s^2) is the rate of change of velocity, whether that change takes place in the direction of motion or not. The unit of acceleration is the acceleration of a body which undergoes unit change of velocity in unit time, or an acceleration of one centimetre per second per second. The acceleration due to gravity is considerably greater than this, for the velocity imparted by gravity to falling bodies in one second is about 981 centimetres per second (or about 32·2 feet per second). The value differs slightly in different latitudes. At Greenwich the value of the acceleration due to gravity is g = 981·17; at the Equator, g = 978·1; and at the North Pole, g = 983·1.

---

* The Author is indebted to his Publishers, Charles Griffin & Co., for liberty to abstract the following pages on this subject from the latest edition of Munro and Jamieson's *Pocket-Book of Electrical Rules and Tables for Electricians and Engineers*, to which the student is referred for further values and definitions.—A. J.

Force (F or f) is that which tends to alter a body's natural state of rest or uniform motion in a straight line.

Force is measured by the rate of change of momentum which it produces, or mass × acceleration.

The *Unit of Force*, or *Dyne*, is that force which, acting for one second on a mass of one gramme, gives to it a velocity of one centimetre per second. The force with which the earth attracts any mass is usually called the "weight" of that mass, and its value obviously differs at different points of the earth's surface. The force with which a body gravitates—*i.e.*, its weight (in dynes), is found by multiplying its mass (in grammes) by the value of g at the particular place where the force is exerted.

Work is the product of a force and the distance through which it acts. The unit of work is the work done in overcoming unit force through unit distance—*i.e.*, in pushing a body through a distance of one centimetre a force of one dyne. It is called the *Erg*. Since the "weight" of 1 gramme is 1×981 or 981 dynes, the work of raising 1 gramme through the height of 1 centimetre against the force of gravity is 981 ergs or g ergs. One kilogramme-metre = 100,000 (g) ergs. One foot-pound = 13,825 (g) ergs = $1 \cdot 356 \times 10^7$ ergs.

Energy is that property which, possessed by a body, gives it the capability of doing work. *Kinetic* energy is the work a body can do in virtue of its motion. *Potential* energy is the work a body can do in virtue of its position. The unit of energy is the *Erg*.

Power or *Activity* is the rate of working. The unit is called the *Watt* (W) = 10^7 ergs per second, or the work done at the rate of 1 *Joule* (J) per second.

One Horse-power (H.P.) = 33,000 ft.-lbs. per minute = 550 ft.-lbs. per second; but, as seen above under *Work*, 1 ft.-lb. = $1 \cdot 350 \times 10^7$ ergs, and, under *Power*, 1 *Watt* = 10^7 ergs per second.

Hence, a *Horse-power* = $550 \times 1 \cdot 356 \times 10^7$ ergs per sec. = 746 watts.

If E = volts, C = amperes, and R = ohms.

Then, $$\text{H.P.} = \frac{E\,C}{746} = \frac{C^2 R}{746} = \frac{E^2}{746\,R}.$$

PRACTICAL ELECTRICAL UNITS.

1. As a **Unit of Resistance** (R), the International **Ohm** (O or ω), which is based upon the ohm equal to 10^9 units of resistance of the C.G.S. system of electro-magnetic units, and is represented by the resistance offered to an unvarying electric current by a column of mercury at the temperature of melting ice, 14·4521 grammes in mass, of a constant cross-sectional area and of the length of 106·3 centimetres.

2. As a **Unit of Current** (C or *c*), the International **Ampere** (A), which is one-tenth of the unit of current of the C.G.S. system of electro-magnetic units, and which is represented sufficiently well for practical use by the unvarying current which, when passed through a solution of nitrate of silver in water, and in accordance with their specifications, deposits silver at the rate of 0·001118 gramme per second.

3. As a **Unit of Electro-motive Force** (E), the International **Volt** (V), which is the E.M.F. that, steadily applied to a conductor whose resistance is one International Ohm, will produce a current of one International Ampere, and which is represented sufficiently well for practical use by $\frac{1000}{1434}$ of the E.M.F. between the poles or electrodes of the voltaic cell known as Clark's cell, at a temperature of 15° Centigrade, and prepared in the manner described in their specification, or by the *new* Weston cell.

4. As the **Unit of Quantity** (Q), the International **Coulomb** ($A \times s$), which is the quantity of electricity transferred by a current of one International Ampere in one second.

5. As the **Unit of Capacity** (K), the International **Farad** (Fd), which is the capacity of a conductor charged to a potential of one International Volt by one International Coulomb of electricity.

6. As a **Unit of Work** the **Joule** (J), or **Watt-second** ($W_P \times s$), which is 10^7 units of work in the C.G.S. system, and which is represented sufficiently well for practical use by the energy expended in one second *in heating* an International Ohm.

7. As the **Unit of Power** (P_W), the International **Watt** (W_P), which is equal to 10^7 units of power in the C.G.S. system, and which is represented sufficiently well for practical use by the work done at the rate of one Joule per second. The **Kilowatt** (Kw.) = 1,000 Watts = $1\frac{1}{3}$ Horse-power.

8. As the **Unit of Induction** (L), the **Henry** (H), which is the induction in the circuit when the E.M.F. induced in this circuit is one International Volt while the inducing current varies at the rate of one ampere per second.

9. The **Board of Trade Commercial Unit of Work** (B.T.U.) is the **Kilowatt-hour** (Kw-hr.) = 1,000 Watt-hours = $1\frac{1}{3}$ H.P. working for *one* hour. Or, say, 10 amperes flowing in a circuit for 1 hour at a pressure of 100 volts.

NOTE.—For further simple explanations, with examples, see the latest edition of Prof. Jamieson's *Manual of Practical Magnetism and Electricity*. Also, see the latest edition of Munro and Jamieson's *Electrical Engineering Pocket-Book*—both published by Charles Griffin & Co., London.

SYMBOLS PROVISIONALLY ADOPTED AT TURIN, SEPTEMBER 13, 1911.

By the International Electrotechnical Commission.

1. Instantaneous values of electrical quantities which vary with the time are to be represented by small letters.

2. Virtual or constant values of electrical quantities to be represented by capital letters.

3. Maximum values of periodic electrical quantities to be represented by capital letters, followed by the subscript "m."

4. Magnetic quantities, constant or variable, to be represented either by capital script, Gothic, heavy-faced (*i.e.*, block clarendon), or any special type.

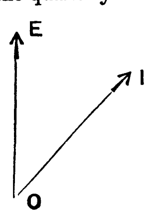
5. Maximum values of magnetic quantities to be represented either by capital script, Gothic, heavy-faced, or any special type, followed by the subscript "m."

6. The following quantities to be represented by the following letters:—

Electromotive force, ...	E, *e*.	
Electric quantity, ...	Q, *q*.	
Inductance,*	**L**.	For type, see pars. 4 and 5.
Magnetic force, ...	**H**.	For type, see pars. 4 and 5.
Magnetic flux density, . . .	**B**.	For type, see pars. 4 and 5.
Length, ...	L, *l*.	
Mass, ...	M, *m*.	
Time, ...	T, *t*.	

The letters I, E, R were definitely adopted to represent the current, electromotive force, and resistance in the simple expression of Ohm's law.

The term "reactive power" was adopted to designate, in A.C. questions, the quantity $UI \sin \phi$.



Diagrams for A.C. Quantities.—In the graphical representation of alternating electric and magnetic quantities, advance in phase shall be represented in the counter-clockwise direction.

NOTE.—In consequence, the impedance of a reactive coil of resistance R and inductance **L** is $R + \sqrt{-1}\,\mathbf{L}\,\omega$, and that of a condenser of capacity C is

$$\frac{1}{\sqrt{-1}\,C\,\omega},$$

where $\omega = 2\pi \times$ frequency. It follows also, that the diagram represents the phase relations in a simple A.C. circuit containing an impressed E.M.F., OE, and a lagging current OI.

* Coefficient of self-induction.

INDEX FOR VOLUME IV.

HYDRAULICS, HYDRAULIC AND REFRIGERATING MACHINERY.

A

ABBREVIATIONS, xv.
Accumulator—Hydraulic, 44.
—— —— Differential, 48.
—— —— Steam, 49.
Air pump, 18.
Ammonia as a refrigerating agent, 206.
Antilogarithms, 220.

B

BEAR, Hydraulic, 41.
Belt-driven suction pump, 15.
Bernouilli's theorem, 94.
Blower, Roots', 31.
Brake, Hoisting, for electric crane, 77.
Bramah's hydraulic press, 32.
Breast wheel, 162.
Brown's steam accumulator, 102.

C

CABLE press, Lead covering, 42.
Capstan, Hydraulic, 56.
Carbon dioxide as a refrigerating agent, 199.
—— —— refrigerating, 204.
Centre of pressure of immersed bodies, 9.
—— —— —— on a circle, 12.
Centre of pressure on a rectangle. 10.
—— —— —— on a triangle, 11.
Centrifugal fans, 185.
—— force, Pressure due to, 128.
—— pumps, 185.
C.G.S. system of units, 255-258.
Circulating pumps, 24.
City and Guilds Exams., General instructions for, 247.
Civil Engineers Exams., Rules and Syllabus of, 223.
Clack Mill, 165.
Coefficient of contraction, 114.
—— of discharge, 117.
—— of friction, 136.
—— of viscosity, 2.
Cold, Transmission of, 214.
Common suction pump, 14.
Comparative trials for efficiency of hydraulic and electric cranes, 82.
Compressor for ammonia, 209.
Continuous delivery pumps, 22.
Cost of hydraulic and electric power, 83.
Crane, Efficiency curves, 88.
—— Explanation of Efficiency Curves, 89.
—— Hydraulic, 46, 70.
—— Movable Electric, 72, 90.
—— —— —— Details of hoisting brake, 77.
—— —— Hydraulic, 70.
—— Tests, 83.
Current meter, 120.

D

DEFINITIONS of units. 255-258.
De la Vergne's double-acting compressor. 209.
—— —— —— refrigerator, 207.
Depth, Hydraulic mean, 135.
Details of hoisting brake and levers for working electric crane, 77.
Differential accumulator, 48.
Direct-acting steam pump, 52.
Diving bell, Pressure in, 8.

E

EFFICIENCY curves of hydraulic and electric cranes, 88, 91.
—— —— Explanation of, 65, 89.
—— of a machine, 62.
—— of a reversible machine, 67.
—— of water turbines, 183.
Ejectors, 118.
Electrical units, 255-258.
Electric cranes, Fixed, 90.
—— —— Movable, 72, 90.
—— power, Cost of, 83.
Elliott's current meter, 120.
Energy lost by impact, 132.
—— of flowing water, 94.
—— of hydraulic accumulator, 48.
—— of still water, 13.

F

FAIRBAIRN'S breast-wheel, 162.
—— improvement, 164.
Fans and pumps, Centrifugal, 185.
Ferris-Pitot meter, 125.
Fixed electric cranes for Manchester Ship Canal, 90.
Flanging press, 35.
Flow of water from tanks and notches, 113.
Fluid, Definition of, 1.
—— pressure, Examples on, 5.
—— Pressure of, 3.
—— Transmission of pressure by a, 3.
Fluids, Impact of solids and, 132.
—— Viscosity of, 2.
Force pumps, 19.
Forced vortex, 128.
Free surface, 3.
—— vortex. 127.
Friction factor, 137.
—— head, 136.
Frictional resistances of machines, 62.
Functions of angles, 222.
Fundamental units, 254.

G

GASES, 1.
Gauge notch, 115.
General Instructions—
City and Guilds Exams., 247.
Civil Engineers Exams., 223.
Giant turbine, Little, 179.
Girard turbine, 170, 187.
—— —— Günther's, 170.
Governor, Günther's turbine, 176.
—— King's turbine, 188.
Günther's Girard turbine, 170.
—— Jonval turbine, 175.
—— turbine governor, 176.

H

HALL'S refrigerator, 204.
Head due to friction in water pipes Loss of, 136.
—— of fluid, 3.
—— Measurement of, 119.
Hercules turbine, 183.
Hoist, Movable jigger, 49.
Horse-power of a stream, 123.
Hydraulic accumulator, 44.
—— —— Differential, 48.
—— —— Steam, 49.
—— bear, 41.
—— capstan, 56.
—— cranes, 46-51.
—— flanging press, 35.
—— jack, 37.
—— Movable, crane, 70.
—— lead-covering cable press, 42.
—— mean depth, 135.
—— motors, 160.
—— power, Cost of, 83.
—— press, Bramah's, 32.
—— ram, 97.

Hydraulics, 1.
Hydrokinetics, 1.
Hydrostatics, 1.

I

IMMERSED surface, Pressure on, 4.
Impact of fluids and solids, 132.
Index letters, xv.
Injectors, 96.
Institution of Civil Engineers, Rules and Syllabus of Exams., 223.
Inward-flow turbines, 177, 189.

J

JACK, Hydraulic, 37.
Jet, Motion produced by, 130.
—— pumps, 96.
Jigger hoist, 49.
Jonval turbine, 175.

K

KENT'S water recorder, 108.
King's turbine governor, 188.

L

LEAD-COVERING cable press, 42.
Level, Slope of free, 135.
Lift of a valve, 20.
Linde refrigerator, 212.
Liquids, 1.
Logarithms, Table of, 218.
Loss of energy by impact, 132.
—— of head due to friction in water pipe, 136.

M

MACHINE, Efficiency of a, 62.
—— Frictional resistance of a, 62.
—— Refrigerating, 200.
—— —— De La Vergne's, 207.
—— —— Hall's, 204.
—— —— Linde, 212.
Manchester Ship Canal, Electric cranes for the, 90.
Measurement of a flowing stream, 115.
—— of head, 119.
Mechanical units, 254.
Meter, Elliott's current, 120.
—— Ferris-Pitot, 125.
—— Venturi, 102.
Mill, Clack, 165.
Mixed-flow turbines, 179, 189.
Momentum of a body, 168.
Motion produced by a jet, 130.
—— Vortex, 126.
Motors, Hydraulic, 160, 297.
Movable jigger hoist, 49.

N

NON-REVERSIBILITY of machines, Conditions for, 69.
Notches, Flow over, 115.

O

ORIFICE, Reduction in pressure round an, 132.
Overshot water wheel, 161.

P

PELTON wheel, 166, 186.
—— —— regulation, 186.
Perimeter of a pipe or channel, Wetted, 135.
Pipes, Resistance flow of fluids, 134.
—— of varying diameter, Discharge of, 141.
Pitot tube, 124.
Power, Cost of hydraulic and electric, 83.
Pressure, Centre of, 9.
—— column, 4.
—— due to centrifugal force, 128.
—— Examples on fluid, 5.
—— in diving bell, 8.
—— of a fluid, 3.
—— on an immersed surface, 4.
—— on sides of tank, 7.

Pressure, Reduction of, round and orifice, 132.
—— Slope of, 135.
—— Transmission of, by a fluid, 3.
Principle of work applied to machines, 63.
—— of work applied to the steam engine, 66.
Pulsometer pumps, 28.
Pumping engines, 97.
Pumps, Air, 18.
—— Centrifugal, 185.
—— Circulating, 24.
—— Continuous delivery, 22.
—— Force, 19.
—— Jet, 96.
—— Pulsometer, 28.
—— Steam, 25.
—— Suction, 14, 16.

R

Ram, Hydraulic, 98.
Rate of change of momentum, 168.
Reaction of a jet, 129.
Recorder, Kent's water, 108.
Rectangle, Centre of pressure on a, 10.
Rectangular gauge notch, 115.
Reduction in pressure round an orifice, 132.
Refrigerating agents—
Anhydrous ammonia, 206.
Carbon dioxide, 199.
Refrigerating apparatus, 199.
—— Elementary, 199.
—— machine, De la Vergne's, 207.
—— —— Hall's, 204.
—— —— Linde, 212.
—— —— Simple, 200.
Refrigeration, 198.
—— Methods of transmitting cold, 214.
Regulation of Pelton wheel, 186.
Resistance of pipes to flow of fluids, 141.
Roots' blower, 31.
Rules and Syllabus of the Institution of Civil Engineers Examinations, 223.

S

Slope of free level and pressure, 135.
Sluice gate, Pressure on, 12.
Sluices, Discharge through, 123.
Solids, Impact of fluids and, 132.
Specification for movable hydraulic crane, 70.
—— for electric crane, 72.
Steam ejector, 96.
—— engine, Application of principle of work to, 66.
—— loaded accumulator, 49.
—— pump, 25.
Still water, Energy of, 13.
Stoneywood Paper Works, turbine installation, 182.
Stream, Horse-power of a, 123.
—— Measurement of a flowing, 115.
Suction pump, 14.
Surface, Pressure on an immersed, 4.
Syllabus and Rules of examinations for election of Associate Members to the Inst.C.E., 223.
Symbols, xv, 54.

T

Tables—
Discharge over ⊔ boards, 122.
—— through sluices, etc., 123.
Friction head of clean pipe, 138.
Functions of angles, 222.
Logarithms, 218-221.
Results of tests of electric crane, 87.
Results of tests of hydraulic crane, 86.
Tests, Crane, 83.
Thomson's triangular gauge notch, 118.
—— vortex turbine, 177.
Transmission of cold, 214.
—— of pressure by fluids, 3.
Triangle, Centre of pressure on a, 11.
Triangular gauge notch, 118.
Turbine governor, Günther's, 176.
—— —— King's, 188.
—— installation at Stoneywood Paper Works, 182.
Turbines, Efficiency, 167.
—— Girard, 170, 187.

Turbines, Hercules, 183.
—— Jonval, 175, 188.
—— Little Giant, 178.
—— Thomson's vortex, 177.
—— Waverley mixed-flow, 189.
Tweddell's differential accumulator, 47.

U

Undershot water wheels, 163.
Units and Definitions, 255-258.
Useful constants, 217.

V

Valve, Lift of a, 19.
Vapours, 1.
Velocity of efflux of water, 113.
Venturi law, tube, and meter, etc., 102.
Viscosity of fluids, 2.
Vortex, Forced, 127.
—— Free, 126.
—— motion, 126.

W

Wall crane, Hydraulic, 46.
Water, Energy of flowing, 94.
—— —— of still, 13.
—— measurement, 115.
—— pipes, Loss of head due to friction in, 134.
—— Pressure of, 3, 13.
—— recorder, Kent's, 108.
—— Velocity of efflux and flow of, from a tank, 113.
Water-wheel, Overshot, 161.
—— Pelton, 166, 186.
—— Undershot, 163.
Water-wheels, 164.
—— Efficiency of, 167.
Waverley mixed-flow turbine, 189.
Weir's discharge over ⊔ boards, 115.
Wetted perimeter of a pipe or channel, 135.
Wheel, Pelton, 166.
Work, Principle applied to machines, 63.
—— —— —— —— steam engines, 66.
Worthington steam pump, 25.

BELL AND BAIN, LIMITED, PRINTERS, GLASGOW